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# Stationary Internal Combustion Engines Standards Support and Environmental Impact Statement. Volume I: Proposed Standards of Performance

(U.S.) Environmental Protection Agency
Research Triangle Park, NC

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Standards of performance to control nitrogen oxides from new, modified and reconstructed stationary internal combustion engines in the U. S. are being proposed under section 111 of the Clean Air Act. This document contains information on the internal combustion engine industry and emission control technology, a discussion of the selected emission limits and the supporting data and the alternatives which were considered, and analyses of the environmental and economic impacts of the proposed standards.

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EPA-450/2-78-125a

# Stationary Internal Combustion Engines

## Standards Support and Environmental Impact Statement Volume I: Proposed Standards of Performance

by

Emission Standards and Engineering Division

U.S. ENVIRONMENTAL PROTECTION AGENCY
Office of Air, Noise, and Radiation
Office of Air Quality Planning and Standards
Research Triangle Park, North Carolina 27711

July 1979

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# 1. SUMMARY

## 1.1 PROPOSED STANDARDS

Standards of performance for stationary internal combustion engines are being proposed under Section 111 of the Clean Air Act. These standards would limit emissions of nitrogen oxides from diesel and dual-fuel stationary internal combustion engines with greater than 560 cubic inch displacement per cylinder (CID/cyl) and gas engines with greater than 350 cubic inch displacement per cylinder or equal to or greater than 8 cylinders and greater than 240 cubic inch displacement per cylinder.

The numerical emission limits for $NO_x$ would be 0.0700 percent by volume (700 ppm) for gas engines, 0.0600 percent by volume (600 ppm) for diesel engines, and 0.0600 percent by volume (600 ppm) for dual-fuel engines corrected to 15 percent oxygen on a dry basis and referenced to standard atmospheric conditions of 29.92 inches mercury, 85 degrees Fahrenheit, and 75 grains moisture per pound of dry air. The proposed standards would also include an adjustment factor for engine efficiency which would adjust the emissions limits upward linearly for IC engines with thermal efficiencies greater than 35 percent. $NO_x$ emissions from stationary internal combustion engines, therefore, would be limited according to one of the following equations:

$$\text{STD} = 700 \left( \frac{5.13}{Y} \text{ kJ/w-hr} \right) \text{for any gas engine}$$

$$\text{STD} = 600 \left( \frac{5.13}{Y} \text{ kJ/w-hr} \right) \text{for any diesel or dual-fuel engine}$$

where:

STD = allowable $NO_x$ emissions (parts per million volume corrected to 15 percent oxygen on a dry basis).

Y = manufacturer's rated brake-specific fuel consumption at peak load (kilojoules per watt hour) or owner/operator's brake-specific fuel consumption at peak load as determined in the field.

During performance tests to determine compliance with the proposed standards, measured $NO_x$ emissions at 15 percent oxygen would be adjusted to standard atmospheric conditions by the appropriate correction factor in the table below:

| Fuel | Correction Factor |
|---|---|
| Diesel and Dual-Fuel | $K = 1/(1 + 0.00235(H - 75) + 0.00220\ (T - 85))$ |
| Gas | $K = (K_H)\ (K_T)$<br>$K_H = 0.844 + 0.151\ (\frac{H}{100}) + 0.075\ (\frac{H}{100})^2$<br>$K_T = 1 - (T - 85)(0.0135)$ |

where:

H = observed humidity, grains $H_2O$/lb dry air

T = observed inlet air temperature, °F

Internal combustion engine manufacturers, owners, or operators may also elect to develop custom ambient condition correction factors, in terms of ambient temperature, and/or humidity, and/or ambient pressure.

These factors must also be substantiated with data and approved for use by EPA.

The proposed standards would apply to facilities that commence construction 30 months after the date of publication in the FEDERAL REGISTER.

Emergency-standby internal combustion engines and all one- and two-cylinder gas engines would be exempt from the $NO_x$ emission limit.

## 1.2 ENVIRONMENTAL/ECONOMIC IMPACT

Four emission control techniques, or combinations of these techniques, have been identified as demonstrated $NO_x$ emission reduction systems for stationary internal combustion engines. These techniques are: (1) retarded ignition or fuel injection, (2) air-to-fuel ratio changes, (3) manifold air cooling, and (4) derating power output (at constant speed). In general, all four techniques are applied by changing an engine operating adjustment.

Fuel injection retard is the most effective $NO_x$ control technique for diesel engines. Similarly, air-to-fuel ratio change is the most effective $NO_x$ control technique for gas engines. Both retard and air-to-fuel ratio changes are effective in reducing $NO_x$ emissions from dual-fuel engines.

Due to technical considerations, ignition retard in excess of eight degrees in diesel engines and changes in the air-to-fuel ratio in excess of five percent in gas engines are the limits to which these techniques could be realistically applied. Eight degrees of ignition retard in diesel engines and a five percent change in air-to-fuel ratios in gas engines yield about a 40 percent reduction in $NO_x$ emissions. Actual emissions varied among engine types; however, the degree of reduction was consistent.

Consequently, a 40 percent reduction in $NO_x$ emissions was the most stringent regulatory option which could be selected as a basis for the standards. An alternative of 20 percent $NO_x$ emission reduction was also considered a viable regulatory option which could serve as the basis for standards of performance.

The main environmental benefit of a standard based on either alternative would be a reduction in national $NO_x$ emissions, which is now 14.6 million megagrams per year for all stationary sources. Total $NO_x$ emissions would decline by 72,500 megagrams annually for alternative I (20 percent reduction) and 145,000 megagrams annually for alternative II (40 percent reduction) in the fifth year after the standard went into effect.

Ambient air quality dispersion modeling based on "worst case" conditions indicates uncontrolled ambient air $NO_x$ levels near large stationary IC engines can vary from approximately 60 percent of the National Ambient Air Quality Standard (NAAQS) of 100 $\mu g/m^3$ to over twice the standard depending on the size of the engine. Thus, standards of performance based on alternative II would be more effective in reducing ambient air $NO_x$ levels than standards of performance based on alternative I.

Emissions of CO and HC, however, would increase, particularly from naturally aspirated gas engines. The magnitude of the increase would be large for CO; however, for both HC and CO, significant emission reductions are readily achievable in other source categories. $NO_x$

emissions, on the other hand, are difficult to reduce, and stationary internal combustion engines offer one of few opportunities for significant $NO_x$ reduction. Therefore, $NO_x$ emissions were selected for control by standards of performance.

There would be essentially no water pollution, solid waste, or noise impact of standards of performance based on either alternative I or alternative II.

The potential energy impact of standards of performance based on either alternative is small. The potential energy impact in the fifth year after the standards go into effect, based on alternative I, would be equivalent to an increase in fuel consumption of approximately 1.0 million barrels of oil per year compared to the IC engine fuel consumption of engines affected by the standards of 31 million barrels per year. The potential energy impact in the fifth year after the standard goes into effect, based on alternative II, would be equivalent to approximately 1.5 million barrels of oil per year. The impact of alternative II represents only 0.02 percent of the 1977 domestic consumption of crude oil and natural gas and only 0.02 percent of the projected total U.S. oil imports five years after the standards go into effect.

Economic impacts on manufacturers or users of stationary internal combustion engines are small. Manufacturers of stationary internal combustion engines would incur additional costs as a result of standards of performance. The manufacturers' total capital investment requirements for developmental testing of engine models is estimated to be about $4.5 million to comply with standards of performance based on alternative I and about \$5 million to comply with standards of performance based on alternative II. Analyses of the financial statements and other public financial information of engine manufacturers indicate that the manufacturers' overhead budgets are sufficient to support the development of these controls without adverse impact on their financial position. Manufacturers would not experience significant differential cost impacts among competing engine model families. Based on "worst-case" assumptions, the maximum intra-industry sales losses would be about six percent as a result of standards of performance based on either alternative. In addition, the impact with regard to increasing sales of gas turbines would be minimal, as a result of standards of performance based on either alternative.

The application of $NO_x$ controls will also increase costs to the engine user. The magnitude of this increase will depend on the amount and type of emission control applied. Fuel penalties are the major factor affecting this increase.

A two percent increase in price would be expected on the average as the result of standards of performance based on either alternative. The total additional capital cost for all users would equal about \$9.6 million on a cumulative basis over the first five years to comply with standards of performance based on either alternative. Total uncontrolled annualized costs of about \$580 million by all large stationary IC engine users would increase, due mainly to fuel penalties, by about \$25 million to comply with standards of performance based on alternative I and would increase by about \$32 million to comply with standards of performance based on alternative II in the fifth year after the standards go into effect.

These impacts translate into price increases for the end products or services provided by the industrial and commercial users of large stationary IC engines. The electric utility industry would pass on a price increase after five years of 0.02 percent to comply with standards of performance based on either alternative. After five years, delivered natural gas prices would increase 0.2 percent as a result of standards of performance based on alternative I and 0.04 percent as a result of standards of performance based on alternative II. Even after a full phase-in period of 30 years, during which new controlled engines would replace all existing uncontrolled engines, these increases would be 0.1 percent for electric utilities, and 0.1 and 0.3 percent for delivered natural gas prices as a result of standards of performance based on alternatives I and II, respectively.

Based on this assessment of the impacts of each alternative, and since alternative II achieves a greater degree of $NO_x$ reduction, it is selected as the best technological system of continuous emission reduction of $NO_x$ from stationary large-bore IC engines, considering the cost of achieving such emission reduction, any nonair quality health and environmental impact, and energy requirements.

## 1.3 INFLATIONARY IMPACT

An economic impact analysis would have to be developed if the proposed standard caused an increase in the fifth-year annualized cost of more than $100 million, a major product price increase of five percent, or an increase in national energy consumption of 25,000 barrels of oil per day. The proposed standard of performance would increase operating costs $26 million in the fifth year, the largest price increases would be approximately two percent, and energy consumption would increase 4,300 barrels of oil per day. The Agency, therefore, feels that no economic impact analysis is required.

# CHAPTER 2

# INTRODUCTION

Standards of performance are proposed following a detailed investigation of air pollution control methods available to the affected industry and the impact of their costs on the industry. This document summarizes the information obtained from such a study. Its purpose is to explain in detail the background and basis of the proposed standards and to facilitate analysis of the proposed standards by interested persons, including those who may not be familiar with the many technical aspects of the industry. To obtain additional copies of this document or the Federal Register notice of proposed standards, write to EPA Library (MD-35), Research Triangle Park, North Carolina 27711. Specify Standards Support and Environmental Impact Statement: Proposed Standards of Performance for Stationary Internal Combustion Engines, report number EPA-450/3-78-125a when ordering.

## 2.1 AUTHORITY FOR THE STANDARDS

Standards of performance for new stationary sources are established under Section 111 of the Clean Air Act (42 U.S.C. 7411), as amended, hereafter referred to as the Act. Section 111 directs the Administrator to establish standards of performance for any category of new stationary source of air pollution which ". . . causes or contributes significantly to, air pollution which may reasonably be anticipated to endanger public health or welfare."

The Act requires that standards of performance for stationary sources reflect, ". . . the degree of emission limitation achievable through the application of the best technological system of continuous emission reduction . . . the Administrator determines has been adequately demonstrated." In addition, for stationary sources whose emissions result from fossil fuel combustion, the standard must also include a percentage reduction in emissions. The Act also provides that the cost of achieving the necessary emission reduction, the nonair quality health and environmental impacts, and the energy requirements all be taken into account in establishing standards of performance. The standards apply only to stationary sources, the construction or modification of which commences after regulations are proposed by publication in the Federal Register.

The 1977 amendments to the Act altered or added numerous provisions which apply to the process of establishing standards of performance.

1. EPA is required to list the categories of major stationary sources which have not already been listed and regulated under standards of performance. Regulations must be promulgated for these new categories on the following schedule:

   25 percent of the listed categories by August 7, 1980
   75 percent of the listed categories by August 7, 1981
   100 percent of the listed categories by August 7, 1982

   A governor of a State may apply to the Administrator to add a category which is not on the list or to revise a standard of performance.

2. EPA is required to review the standards of performance every 4 years, and if appropriate, revise them.
3. EPA is authorized to promulgate a design, equipment, work practice, or operational standard when an emission standard is not feasible.
4. The term "standards of performance" is redefined and a new term "technological system of continuous emission reduction" is defined. The new definitions clarify that the control system must be continuous and may include a low-polluting or nonpolluting process or operation.
5. The time between the proposal and promulgation of a standard under Section 111 of the Act is extended to 6 months.

Standards of performance, by themselves, do not guarantee protection of health or welfare because they are not designed to achieve any specific air quality levels. Rather, they are designed to reflect the degree of emission limitation achievable through application of the best adequately demonstrated technological system of continuous emission reduction, taking into consideration the cost of achieving such emission reduction, any nonair quality health and environmental impact and energy requirements.

Congress had several reasons for including these requirements. First, standards with a degree of uniformity are needed to avoid situations where some States may attract industries by relaxing standards relative to other States. Second, stringent standards enhance the potential for long-term growth. Third, stringent standards may help achieve long-term cost savings by avoiding the need for more expensive retrofitting when pollution ceilings may be reduced in the future.

Fourth, certain types of standards for coal burning sources can adversely affect the coal market by driving up the price of low-sulfur coal or effectively excluding certain coals from the reserve base because their untreated pollution potentials are high. Congress does not intend that new source performance standards contribute to these problems. Fifth, the standard-setting process should create incentives for improved technology.

Promulgation of standards of performance does not prevent State or local agencies from adopting more stringent emission limitations for the same sources. States are free under Section 116 of the Act to establish even more stringent emission limits than those established under Section 111 or those necessary to attain or maintain the national ambient air quality standards (NAAQS) under Section 110. Thus, new sources may in some cases be subject to limitations more stringent than standards of performance under Section 111, and prospective owners and operators of new sources should be aware of this possibility in planning for such facilities.

A similar situation may arise when a major emitting facility is to be constructed in a geographic area which falls under the prevention of significant deterioration of air quality provisions of Part C of the Act. These provisions require, among other things, that major emitting facilities to be constructed in such areas are to be subject to best available control technology. The term "best available control technology" (BACT), as defined in the Act, means ". . . an emission limitation based on the maximum degree of reduction of each pollutant subject to regulation under this Act emitted from or which results from any major emitting facility, which the permitting authority, on a case-by-case basis, taking into account energy, environmental, and economic impacts and other costs, determines is achievable for such facility through application of production processes and available methods, systems, and techniques, including fuel cleaning or treatment or innovative fuel combustion techniques for control of each such pollutant. In no event shall application of best available control technology result in emissions of any pollutants which will exceed the emissions allowed by any applicable standard established pursuant to Section 111 or 112 of this Act."

Although standards of performance are normally structured in terms of numerical emission limits where feasible, alternative approaches are sometimes necessary. In some cases physical measurement of emissions from a new source may be impractical or exorbitantly expensive. Section 111(h) provides that the Administrator may promulgate a design or equipment standard in those cases where it is not feasible to prescribe or enforce a standard of performance. For example, emissions of hydrocarbons from storage vessels for petroleum liquids are greatest during tank filling. The nature of the emissions, high concentrations for short periods during filling, and low concentrations for longer periods during storage, and the configuration of storage tanks make direct emission measurement impractical. Therefore, a more practical approach to standards of performance for storage vessels has been equipment specification.

In addition, Section 111(h) authorizes the Administrator to grant waivers of compliance to permit a source to use innovative continuous emission control technology. In order to grant the waiver, the Administrator must find: (1) a substantial likelihood that the technology will produce greater emission reductions than the standards require, or an equivalent reduction at lower economic, energy or environmental cost;

(2) the proposed system has not been adequately demonstrated; (3) the technology will not cause or contribute to an unreasonable risk to public health, welfare or safety; (4) the governor of the State where the source is located consents; and that, (5) the waiver will not prevent the attainment or maintenance of any ambient standard. A waiver may have conditions attached to assure the source will not prevent attainment of any NAAQS. Any such condition will have the force of a performance standard. Finally, waivers have definite end dates and may be terminated earlier if the conditions are not met or if the system fails to perform as expected. In such a case, the source may be given up to 3 years to meet the standards, with a mandatory progress schedule.

## 2.2 SELECTION OF CATEGORIES OF STATIONARY SOURCES

Section 111 of the Act directs the Administrator to list categories of stationary sources which have not been listed before. The Administrator, ". . . shall include a category of sources in such list if in his judgment it causes, or contributes significantly to, air pollution which may reasonably be anticipated to endanger public health or welfare." Proposal and promulgation of standards of performance are to follow while adhering to the schedule referred to earlier.

Since passage of the Clean Air Amendments of 1970, considerable attention has been given to the development of a system for assigning priorities to various source categories. The approach specifies areas of interest by considering the broad strategy of the Agency for implementing the Clean Air Act. Often, these "areas" are actually pollutants which are emitted by stationary sources. Source categories which emit these pollutants were then evaluated and ranked by a process involving such factors as: (1) the level of emission control (if any) already required by

State regulations; (2) estimated levels of control that might be required from standards of performance for the source category; (3) projections of growth and replacement of existing facilities for the source category; and (4) the estimated incremental amount of air pollution that could be prevented, in a preselected future year, by standards of performance for the source category. Sources for which new source performance standards were promulgated or are under development during 1977 or earlier, were selected on these criteria.

The Act amendments of August 1977, establish specific criteria to be used in determining priorities for all source categories not yet listed by EPA. These are:

1. The quality of air pollutant emissions which each such category will emit, or will be designed to emit;
2. The extent to which each such pollutant may reasonably be anticipated to endanger public health or welfare; and
3. The mobility and competitive nature of each such category of sources and the consequent need for nationally applicable new source standards of performance.

In some cases, it may not be feasible to immediately develop a standard for a source category with a high priority. This might happen when a program of research is needed to develop control techniques or because techniques for sampling and measuring emissions may require refinement. In the developing of standards, differences in the time required to complete the necessary investigation for different source categories must also be considered. For example, substantially more time may be necessary if numerous pollutants must be investigated from a single source category. Further, even late in the development process, the schedule for completion of a standard may change. For example, inability to obtain emission data from well controlled sources in time to pursue the development process in a systematic fashion may force a change in scheduling. Nevertheless, priority ranking is, and will continue to be, used to establish the order in which projects are initiated and resources assigned.

After the source category has been chosen, determining the types of facilities within the source category to which the standard will apply must be decided. A source category may have several facilities that cause air pollution, and emissions from some of these facilities may be insignificant or very expensive to control. Economic studies of the source category and of applicable control technology may show that air pollution control is better served by applying standards to the more severe pollution sources. For this reason, and because there be no adequately demonstrated system for controlling emissions from certain facilities, standards often do not apply to all facilities at a source. For the same reasons, the standards may not apply to all air pollutants emitted. Thus, although a source category may be selected to be covered by a standard of performance, not all pollutants or facilities within that source category may be covered by the standards.

## 2.3 PROCEDURE FOR DEVELOPMENT OF STANDARDS OF PERFORMANCE

Standards of performance must: (1) realistically reflect best demonstrated control practice; (2) adequately consider the cost, and the nonair quality health and environmental impacts and energy requirements of such control; (3) be applicable to existing sources that are modified or reconstructed as well as new installations; and (4) meet these conditions for all variations of operating conditions being considered anywhere in the country.

The objective of a program for development of standards is to identify the best technological system of continuous emission reduction which has been adequately demonstrated. The legislative history of Section 111 and various court decisions make clear that the Administrator's judgment of what is adequately demonstrated is not limited to systems that are in actual routine use. The search may include a technical assessment of control systems which have been adequately demonstrated but for which there is limited operational experience. In most cases, determination of the ". . . degree of emission reduction achievable ..." is based on results of tests of emissions from well controlled existing sources. At times, this has required the investigation and measurement of emissions from control systems found in other industrialized countries that have developed more effective systems of control than those available in the United States.

Since the best demonstrated systems of emission reduction may not be in widespread use, the data base upon which standards are developed may be somewhat limited. Test data on existing well controlled sources are obvious starting points in developing emission limits for new sources. However, since the control of existing sources generally represents retrofit technology or was originally designed to meet an existing State or local regulation, new sources may be able to meet more stringent emission standards. Accordingly, other information must be considered before a judgment can be made as to the level at which the emission standard should be set.

A process for the development of a standard has evolved which takes into account the following considerations:

1. Emissions from existing well controlled sources as measured
2. Data on emissions from such sources are assessed with consideration of such factors as: (a) how representative the tested source is in regard to feedstock, operation, size, age, etc.; (b) age and maintenance of the control equipment tested; (c) design uncertainties of control equipment being considered; and (d) the degree of uncertainty that new sources will be able to achieve similar levels of control.
3. Information from pilot and prototype installations, guarantees by vendors of control equipment, unconstructed but contracted projects, foreign technology, and published literature are also considered during the standard development process. This is especially important for sources where "emerging" technology appears to be a significant alternative.
4. Where possible, standards are developed which permit the use of more than one control technique or licensed process.
5. Where possible, standards are developed to encourage or permit the use of process modifications or new processes as a method of control rather than "add-on" systems of air pollution control.
6. In appropriate cases, standards are developed to permit the use of systems capable of controlling more than one pollutant. As an example, a scrubber can remove both gaseous and particulate emissions, but an electrostatic precipitator is specific to particulate matter.

7. Where appropriate, standards for visible emissions are developed in conjunction with concentration/mass emission standards. The opacity standard is established at a level that will require proper operation and maintenance of the emission control system installed to meet the concentration/mass standard on a day-to-day basis. In some cases, however, it is not possible to develop concentration/mass standards, such as with fugitive sources of emissions. In these cases, only opacity standards may be developed to limit emissions.

## 2.4 CONSIDERATION OF COSTS

Section 317 of the Act requires, among other things, an economic impact assessment with respect to any standard of performance established under Section 111 of the Act. The assessment is required to contain an analysis of:

(1) the costs of compliance with the regulation and standard including the extent to which the cost of compliance varies depending on the effective date of the standard or regulation and the development of less expensive or more efficient methods of compliance;

(2) the potential inflationary recessionary effects of the standard or regulation;

(3) the effects on competition of the standard or regulation with respect to small business;

(4) the effects of the standard or regulation on consumer cost, and,

(5) the effects of the standard or regulation on energy use.

Section 317 requires that the economic impact assessment be as extensive as practicable, taking into account the time and resources available to EPA.

The economic impact of a proposed standard upon an industry is usually addressed both in absolute terms and by comparison with the control costs that would be incurred as a result of compliance with typical existing State control regulations. An incremental approach is taken since both new and existing plants would be required to comply with State regulations in the absence of a Federal standard of performance. This approach requires a detailed analysis of the impact upon the industry resulting from the cost differential that exists between a standard of performance and the typical State standard.

The costs for control of air pollutants are not the only costs considered. Total environmental costs for control of water pollutants as well as air pollutants are analyzed wherever possible.

A thorough study of the profitability and price-setting mechanisms of the industry is essential to the analysis so that an accurate estimate of potential adverse economic impacts can be made. It is also essential to know the capital requirements placed on plants in the absence of Federal standards of performance so that the additional capital requirements necessitated by these standards can be placed in the proper perspective. Finally, it is necessary to recognize any constraints on capital availability within an industry, as this factor also influences the ability of new plants to generate the capital required for installation of additional control equipment needed to meet the standards of performance.

## 2.5 CONSIDERATION OF ENVIRONMENTAL IMPACTS

Section 102(2)(C) of the National Environmental Policy Act (NEPA) of 1969 requires Federal agencies to prepare detailed environmental impact statements on proposals for legislation and other major Federal actions significantly affecting the quality of the human environment. The objective of NEPA is to build into the decision-making process of Federal agencies a careful consideration of all environmental aspects of proposed actions.

In a number of legal challenges to standards of performance for various industries, the Federal Courts of Appeals have held that environmental impact statements need not be prepared by the Agency for proposed actions under Section 111 of the Clean Air Act. Essentially, the Federal Courts of Appeals have determined that ". . . the best system of emission reduction, . . . require(s) the Administrator to take into account counter-productive environmental effects of a proposed standard, as well as economic costs to the industry . . ." On this basis, therefore, the Courts ". . . established a narrow exemption from NEPA for EPA determination under Section 111."

In addition to these judicial determinations, the Energy Supply and Environmental Coordination Act (ESECA) of 1974 (PL-93-319) specifically exempted proposed actions under the Clean Air Act from NEPA requirements. According to Section 7(c)(1), "No action taken under the Clean Air Act shall be deemed a major Federal action significantly affecting the quality of the human environment within the meaning of the National Environmental Policy Act of 1969."

The Agency has concluded, however, that the preparation of environmental impact statements could have beneficial effects on certain regulatory actions. Consequently, while not legally required to do so by Section 102(2)(C) of NEPA, environmental impact statements will be prepared for various regulatory actions, including standards of performance developed under Section 111 of the Act. This voluntary preparation of environmental impact statements, however, in no way legally subjects the Agency to NEPA requirements.

To implement this policy, a separate section is incuded in this document which is devoted solely to an analysis of the potential environmental impacts associated with the proposed standards. Both adverse and beneficial impacts in such areas as air and water pollution, increased solid waste disposal, and increased energy consumption are identified and discussed.

## 2.6 IMPACT ON EXISTING SOURCES

Section 111 of the Act defines a new source as ". . . any stationary source, the construction or modification of which is commenced . . ." after the proposed standards are published. An existing source becomes a new source if the source is modified or is reconstructed. Both modification and reconstruction are defined in amendments to the general provisions of Subpart A of 40 CFR Part 60 which were promulgated in the Federal Register on December 16, 1975 (40 FR 58416). Any physical or operational change to an existing facility which results in an increase in the emission rate of any pollutant for which a standard applies is considered a modification. Reconstruction, on the other hand, means the replacement of components of an existing facility to the extent that the fixed capital cost exceeds 50 percent of the cost of constructing a comparable entirely new source and that it be technically and economically feasible to meet the applicable standards. In such cases, reconstruction is equivalent to new construction.

Promulgation of a standard of performance requires States to establish standards of performance for existing sources in the same industry under Section 111(d) of the Act if the standard for new sources limits emissions of a designated pollutant (i.e., a pollutant for which air quality criteria have not been issued under Section 108 or which has not been listed as a hazardous pollutant under Section 112). If a State does not act, EPA must establish such standards. General provisions outlining procedures for control of existing sources under Section 111(d) were promulgated on November 17, 1975, as Subpart B of 40 CFR Part 60 (40 FR 53340).

## 2.7 REVISION OF STANDARDS OF PERFORMANCE

Congress was aware that the level of air pollution control achievable by any industry may improve with technological advances. Accordingly, Section 111 of the act provides that the Administrator ". . . shall, at least every 4 years, review and, if appropriate, revise . . ." the standards. Revisions are made to assure that the standards continue to reflect the best systems that become available in the future. Such revisions will not be retroactive but will apply to stationary sources constructed or modified after the proposal of the revised standards.

## CHAPTER 3

# THE STATIONARY RECIPROCATING INTERNAL COMBUSTION ENGINE PROCESS AND INDUSTRY

## 3.1 GENERAL

Stationary reciprocating internal combustion (IC) engines operate on the same principles as the common automobile or truck engine. They can be installed almost anywhere, since they can be instrumented for remote operation, can use gasoline, diesel fuel, natural gas, sewage gas, and certain mixtures of these fuels, and require relatively little water. Engines are manufactured in sizes ranging from less than 1 hp to nearly 50,000 hp, although the largest one built in the United States is 13,500 hp. Installations characteristically have a low physical profile (low buildings, short stacks, little visible emissions, and quiet operation when properly muffled), at least when installed as single units. They are often located in, or adjacent to, large urban centers, where power demands are greatest and pollution problems are often the most severe.

IC engines are being used in a multitude of applications because of their short construction time, ease of installation, and remote operation capability with a variety of fuels over a large range of speeds and loads. These applications range from driving large municipal electrical generators to powering small air compressors and welders. Further details about the manufacturers and users of IC engines are presented in the following subsections.

### 3.1.1 Engine Manufacturers

Approximately 40 firms manufacture IC engines for stationary applications in the United States, and many others produce forgings, fuel systems, turbochargers, heat exchangers, and related components. In addition, many Original Equipment Manufacturers (OEM's) purchase engines to incorporate into such final products as trucks, tractors, compressors, welders, pumps, generator sets, and other machinery. These OEM's are significant customers for some manufacturers, particularly those who make truck and tractor engines. One such manufacturer has stated that he sells to about 400 OEM's, whereas another claims his company deals with 40(1). In addition, several of the manufacturers of small- and medium-sized engines sell their product to dealers and distributors, who, in turn, sell them to the end users. Therefore, the manufacturing companies have little knowledge or control over the end-use of their engines. To further complicate the situation, some engine manufacturers are themselves OEM's; they buy engines from other manufacturers in size ranges that they do not produce, and then mate these purchased engines with their own compressors, generators, etc.

For the purposes of this discussion, manufacturers of IC engines can be categorized into four major groups: (1) firms manufacturing large-bore (greater than 8 inches), low- and medium-speed (less than 1200 rpm) engines; (2) firms manufacturing principally small-bore (less than 6-1/2 inches), medium-power, high-speed engines (greater than 1200 rpm); (3) firms manufacturing low-power (less than 100 hp), high-speed engines and generator sets; and (4) firms which manufacture small (less than 20 hp) one-cylinder, air-cooled gasoline engines principally used for lawn and garden equipment. Table 3-1(2) shows these firms in their respective categories. Although there is some overlap in power ratings offered by members of the first three

## TABLE 3-1. IC ENGINE MANUFACTURERS(2)

| Manufacturer | Bore, inches | CID/cyl[a] | Power Range, hp[b] | Cylinder Power, hp/cyl | Speed Range, rpm |
|---|---|---|---|---|---|
| Large Bore Engines | | | | | |
| Alco | 9 | 666 | 1000 - 4400 | 175 - 250 | 400-1200 |
| Colt | 8 - 16 | 1037-3526 | 850 - 9400 | 175 - 500 | 500-900 |
| Cooper-Bessemer | 13 - 20 | 2155-6283 | 900 - 13500 | 360 - 675 | 250-600 |
| ElectroMotive Div (GM) | 9 | 645 | 800 - 3600 | 100 - 180 | 900 |
| Enterprise | 13 - 17 | 1200-4770 | 1600 - 13500 | 280 - 680 | 400-600 |
| Dresser-Clark | 17 - 19 | 3860-5100 | 1000 - 10000 | 200 - 500 | 300-330 |
| Ingersoll-Rand | 11 - 17 | 1350-4993 | 1000 - 5500 | 125 - 330 | 300-550 |
| Worthington[c] | 14 - 16 | 2972-4021 | 2300 - 8600 | 125 - 330 | 300-500 |
| White Superior | 8 - 14 | 510-825 | 400 - 2650 | 75 - 150 | 900-1000 |
| Medium Bore Engines | | | | | |
| Allis-Chalmers | 3-1/2 - 6 | 43-169 | 29 - 850 | 10 - 70 | 1200-2600 |
| Case | 3-3/4 - 4-1/8 | 47-84 | 50 - 125 | 12 - 30 | 1200-2200 |
| Caterpillar | 4-1/2 - 6-1/4 | 79-246 | 85 - 1000 | 21 - 63 | 1200-2400 |
| Chevrolet-Oldsmobile (GM) | 3-7/8 - 4-1/4 | 26-57 | 50 - 215 | 13 - 27 | 3600-4000 |
| Chrysler | 3-1/4 - 4-1/4 | 28-52 | 43 - 175 | 12 - 30 | 1200-4000 |
| Cooper-Penjax[d] | 5 - 15 | 128-2827 | 15 - 300 | 15 - 150 | 200-900 |
| Cummins | 4-1/2 - 6-1/4 | 63-192 | 120 - 1200 | 20 - 100 | 1200-2100 |
| Detroit Diesel | 3-7/8 - 5-3/4 | 53-149 | 50 - 1100 | 20 - 68 | 1800-2500 |
| Ford | 4-1/8 - 4-1/2 | 26-67 | 38 - 200 | 9 - 22 | 2500-4600 |
| International Harvester | 3-7/8 - 5-3/8 | 39-136 | 16 - 325 | 9 - 50 | 1800-3000 |
| John Deere | 4 - 4-3/4 | 55-89 | 44 - 180 | 15 - 30 | 1500-2500 |
| Minneapolis-Moline | 4-1/4 - 5-5/16 | 71-133 | 90 - 180 | 15 - 29 | 1800 |
| Murphy | 5-1/2 - 6-3/8 | 142-207 | 110 - 520 | 30 - 60 | 1200-1800 |
| Sterling | 3 - 5-1/2 | 16-166 | 16 - 152 | 8 - 20 | 1200-1800 |
| Stewart and Stevenson | 3-7/8 - 4-3/4 | 53-149 | 30 - 1100 | 20 - 68 | 1200-1800 |
| Teledyne Continental | 3-7/8 | 28-41 | 50 - 80 | 12 - 14 | 2000-2400 |
| Waukesha[e] | 3-5/8 - 9-3/8 | 38-586 | 52 - 1550 | 8 - 110 | 1000-1800 |
| White Engines (Hercules) | 3-3/4 - 4-1/2 | 35-80 | 34 - 130 | 12 - 26 | 2400-2800 |

[a] Cubic inch displacement per cylinder

[b] Horsepower is for rated conditions (continuous operation) @ 130°F intercooler water temperature for large-bore engines and 85°F inlet air temperature for medium-bore engines.

[c] Worthington ceased producing engines during the writing of this report.

[d] This manufacturer produces high power one- and two-cylinder engines.

[e] The manufacturer straddles the medium- to large-bore categories, however, the majority of engine production is in the medium-bore category.

## TABLE 3-1. Concluded

| Manufacturer | Bore, inches | CID/cyl[a] | Power Range, hp[b] | Cylinder Power, hp/cyl | Speed Range, rpm |
|---|---|---|---|---|---|
| Small Engines and Generator Sets | | | | | |
| Kohler | — | | 3-3/4 - 28 | 3 - 7 | 2000-2700 |
| Onan | 3-1/4 | | 8 - 30 | 6 - 8 | 1800-3900 |
| Teledyne Wisconsin | 2-3/4 - 4-1/8 | | 3-1/2 - 80 | 3-1/2 - 20 | 2400-3600 |
| Wills Industries | — | | 11 - 28 | 11 - 14 | 3000 |
| Witte | 4-1/4 - 5 | | 9 - 27 | 9 - 14 | 800-1800 |
| Very Small Engines | | | | | |
| Briggs and Stratton | 2-3/8 - 3-9/16 | | 2 - 16 | 2 - 16 | 3100-4000 |
| Chrysler | 2 - 2-1/4 | | 3-1/4 - 8 | 3-1/4 - 8 | 5500-7000 |
| Clinton | 2-1/2 - 2-3/8 | | 4 - 7 | 4 - 7 | 4600-5800 |
| Homelite | 1-7/16 - 2-3/4 | | 2 - 4 | 2 - 4 | — |
| Jacobsen | 2-1/8 | | 3 | 3 | 3600 |
| Outboard Marine Corp. (Lawn Boy) | 2-3/8 | | <10 | — | — |
| McCulloch | 1-3/8 - 2-1/4 | | <20 | — | — |
| O&R | 1-1/8 - 1-17/32 | | 1 - 2-1/4 | 1 - 2-1/2 | 6300-7200 |
| Tecumseh | 2-3/8 - 3-1/2 | | 2-1/2 - 16 | 2-1/2 - 16 | 2500-3600 |

T-660

[a] Cubic inch displacement per cylinder

[b] Horsepower is for rated conditions (continuous operation) @ 130°F intercooler water temperature for large-bore engines and 85°F inlet air temperature for medium-bore engines.

groups (Figure 3-1), the differences tend to be more distinct when viewed on a power-per-cylinder, displacement-per-cylinder, or bore basis (Figures 3-2, 3-3, and 3-4). The latter two show the clearest division between medium- and high-power engines, with only Waukesha and Cooper-Penjax straddling the two categories. More significant, however, is the fact that the market interactions between firms in the different groups are of far less importance than the interactions within the groups. Thus, the stationary IC engine industry is in many ways four industries, each with its own markets and problems.

Any IC engine can be adapted for use in stationary applications (for example, automotive engines are easily adapted for irrigation pumps). Thus, any engine manufacturer is a potential source of stationary engines. However, the following discussion will concentrate only on those manufacturers who market engines specifically for stationary applications.

All but three of the manufacturers are divisions or subsidiaries of large, diversified corporations with stationary engine sales accounting for a small part of total corporate revenue.1/ Total sales of stationary engines are in the range of $300 million to $400 million per year. Since most firms manufacture engines for nonstationary as well as stationary applications, it is difficult to determine employment directly attributable to stationary engines. Assuming, however, that the percentage of employment at a plant attributable to stationary engines is equal to the percentage of sales attributable to stationary engines, it is estimated that between 15,000

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1/The discussion in this paragraph excludes manufacturers of engines under 20 hp. Their annual sales are also in the range of $300 million to $500 million.

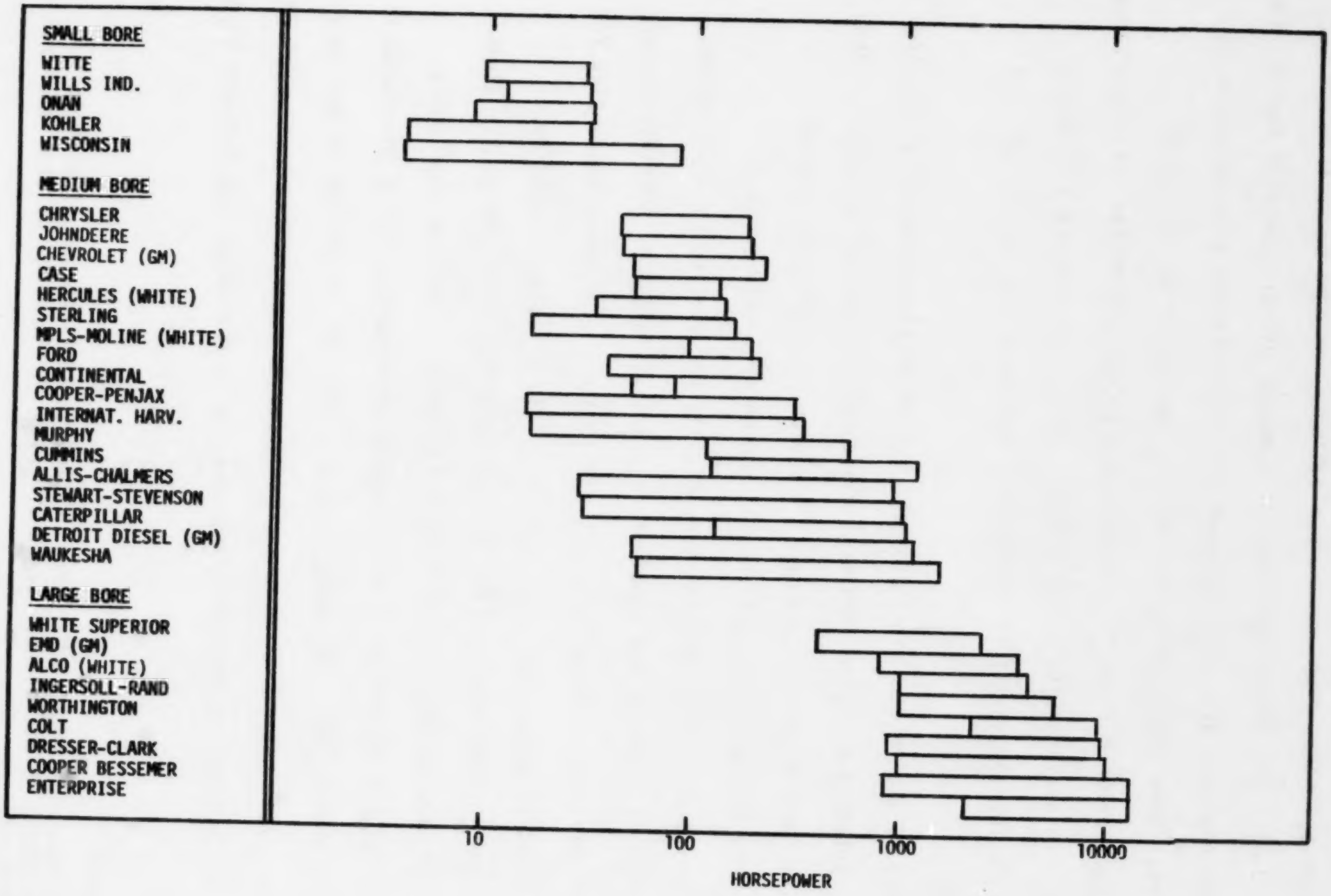

Figure 3-1. Manufacturers categorized by engine horsepower.

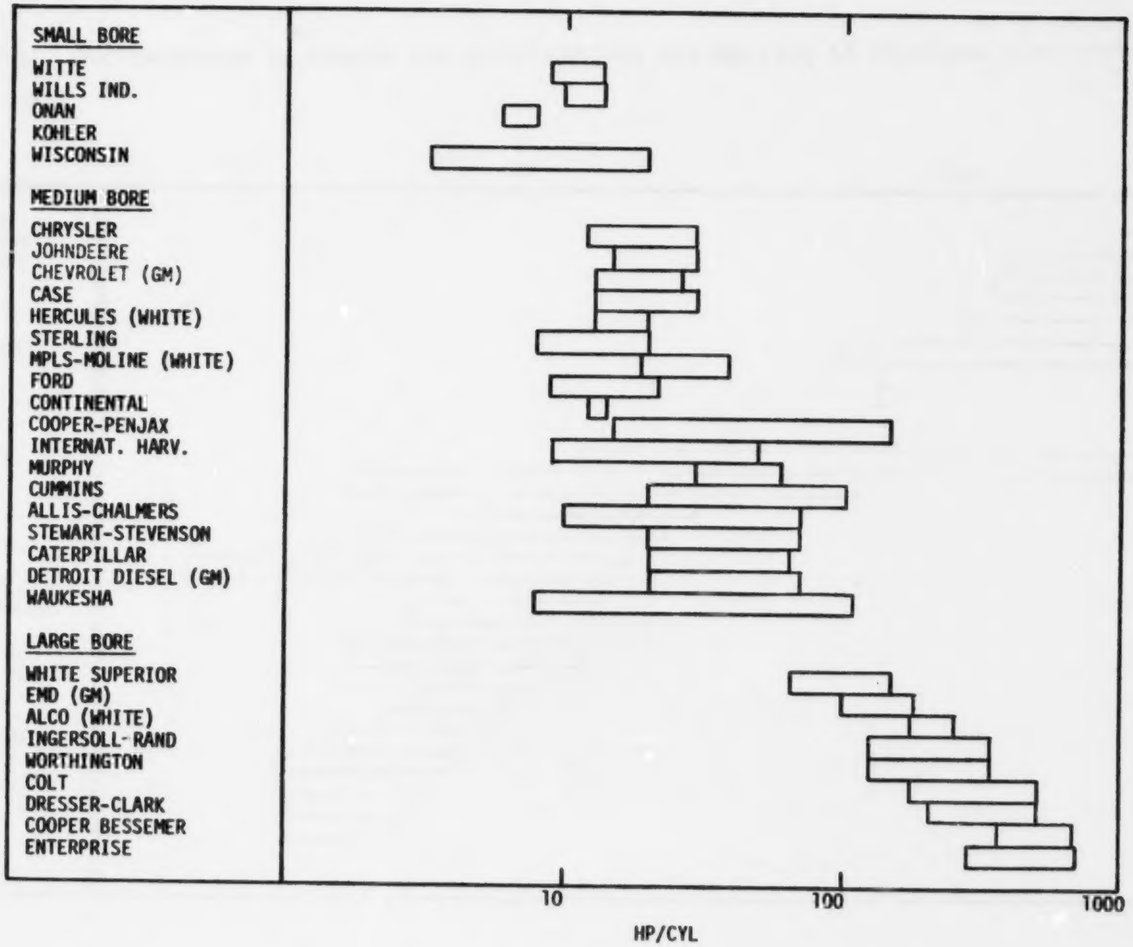

Figure 3-2. Manufacturers categorized by engine horsepower per cylinder.

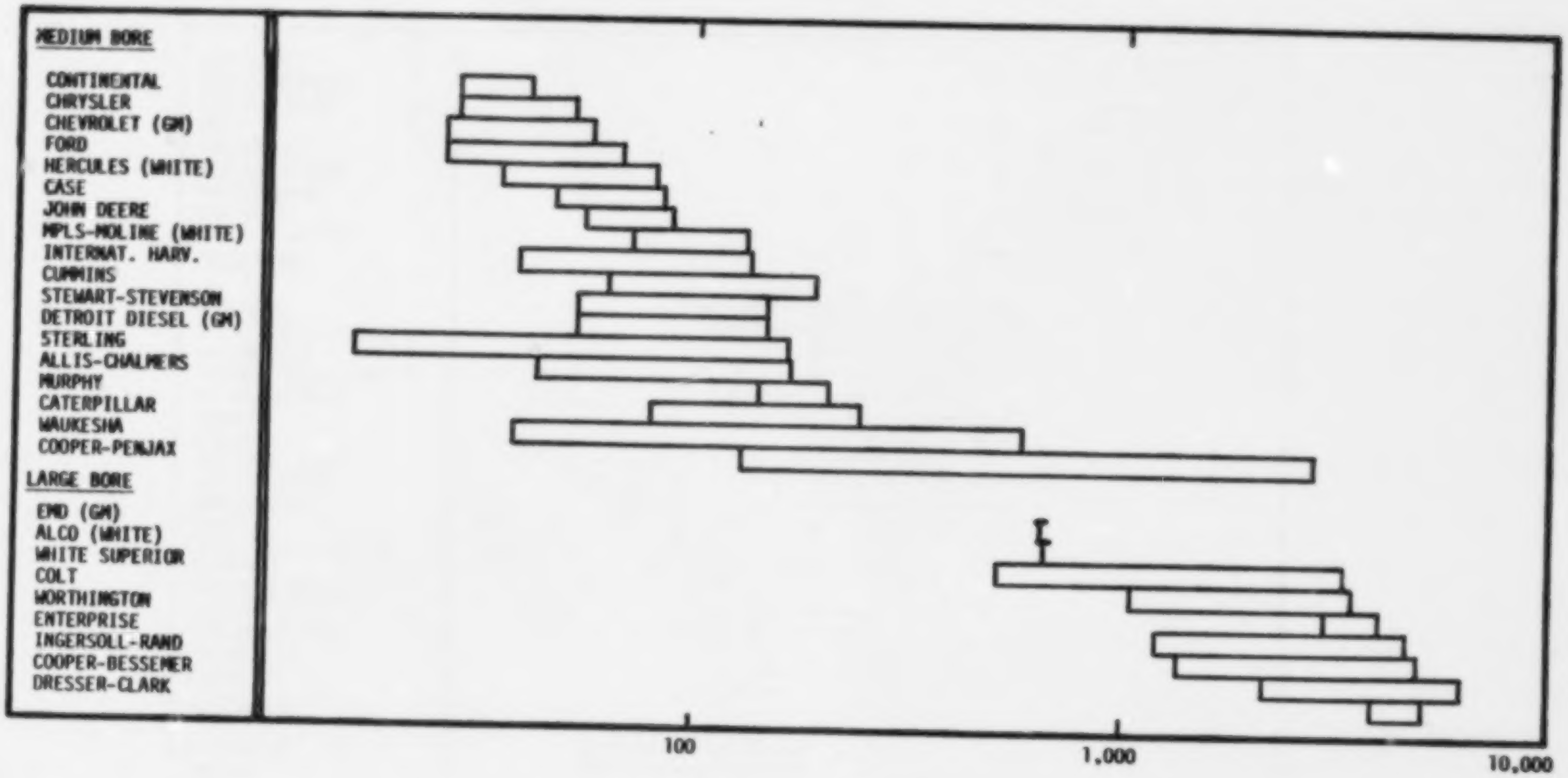

Figure 3-3. Manufacturers of medium and large engines categorized by cylinder displacement, in³/cyl.

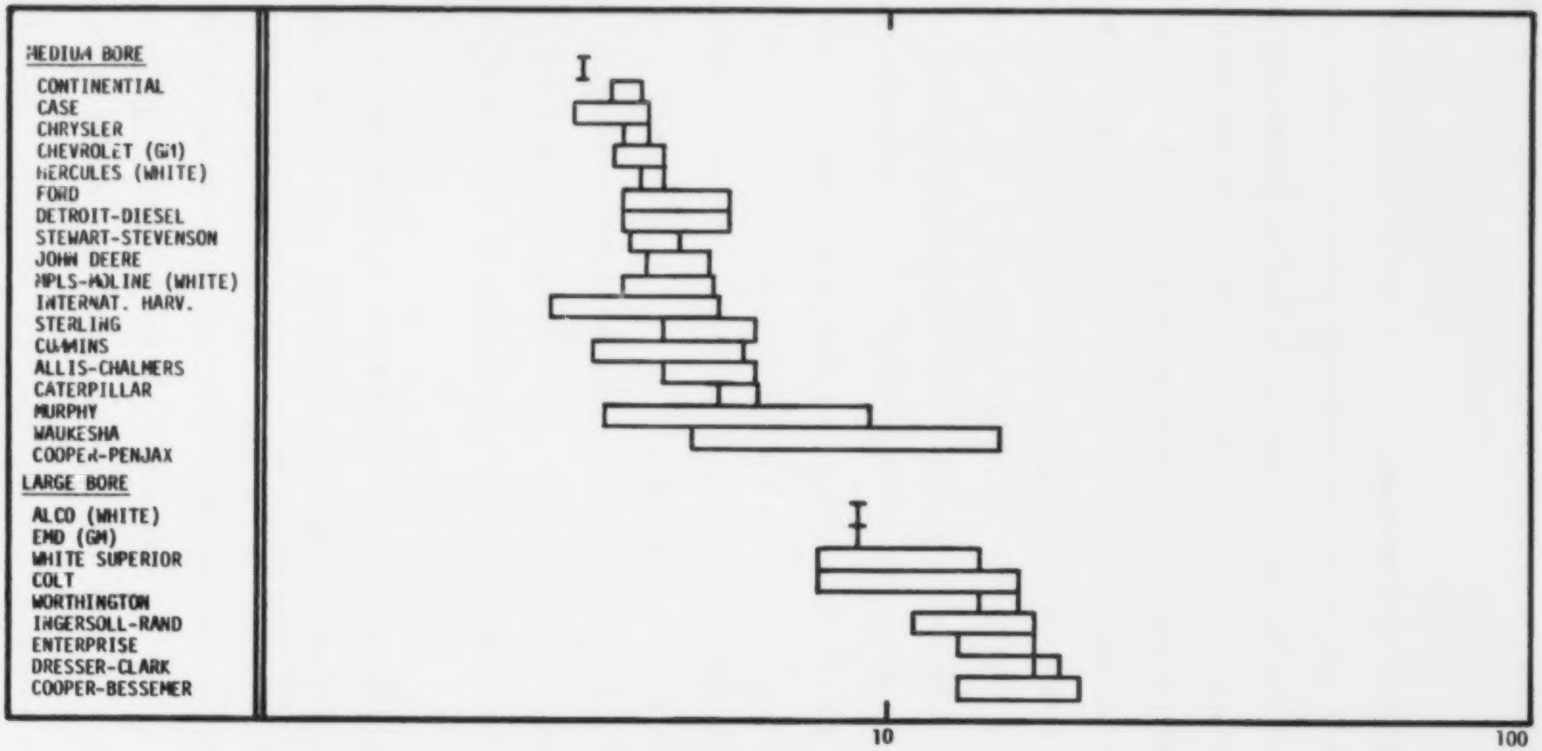

Figure 3-4. Manufacturers of medium and large engines categorized by bore size, in.

and 30,000 people are employed directly in the manufacture of stationary engines.

### 3.1.1.1 Large-Bore, High-Power, Low- and Medium-Speed Engines

Most of the large-bore, high-power (>100 hp/cylinder) engines manufactured are four-cycle, compression-ignition engines designed to operate on either diesel oil or a mixture of oil and natural gas (dual fuel). The remainder of the engines are spark-ignited natural gas engines, about equally distributed between two- and four-cycle varieties. Some engines are built to operate as either spark-ignited or compression-ignited (dual fuel or oil) and can easily be switched in the field in response to fuel availability. Production of large-bore engines for stationary applications is currently in the range of 1000 to 2000 units/year with a total production value of $80 million to $150 million. Sales have been largely constant over the last 5 years, with some firms showing a slight growth and others, particularly those manufacturing gas engines, having declining sales. These manufacturers expect sales to grow at an annual rate of 5 to 10 percent, on a total horsepower basis, for the next 5 years as demand increases for engines for nuclear standby and natural gas pipelines (see Section 7.1.2). Between 5000 and 6000 people are employed in the manufacture of stationary high-power engines.

As Figure 3-1 indicates, there is a region of overlap between the power ranges offered by the manufacturers of large- and medium-bore engines. There are basic differences, however, which separate the two groups. Large-bore engines produce high-power output at low speeds due to their large displacement and consequent high power per cylinder. Smaller bore engines, in contrast, have lower power per cylinder (and therefore more cylinders for the same engine horsepower) and achieve high outputs by utilizing high rotative speeds. Thus, for the same power rating, high-speed engines are smaller, less expensive, and capable of running at a wider range of speeds. Low-speed, large-bore engines, on the other hand, are more economical to operate continuously because of lower fuel consumption and longer lifetimes. For these reasons, high-speed engines tend to find different applications (see below) than low-speed engines of the same power, and there is little competition between firms in these two groups.

### 3.1.1.2 Medium-Power, High-Speed Engines

The firms manufacturing medium-size, high-speed engines (110 to 100 hp/cylinder and greater than 1000 rpm) vary greatly in size, employing from less than 200 to more than 10,000 people in engine manufacturing. Although four firms, Cummins, Caterpillar, Detroit Diesel Allison Division of General Motors, and Ford, dominate diesel engine production (with annual sales of 60,000 to 120,000 units each), they concentrate their sales in the truck, tractor, construction, and material handling markets(3). Ford, General Motors, and International Harvester also manufacture significant quantities of gasoline-fueled truck and tractor engines. Therefore, most of the stationary engines produced by these manufacturers are mobile engines modified for stationary installation and constant speed operation. Three additional firms, International Harvester, Waukesha, and Teledyne Continental, each sell about as many engines for stationary applications (5000 to 15,000 per year) as do the four larger companies(4,5,6). The remaining manufacturers in this group are smaller.

Medium-power engines are predominantly gasoline- and diesel-fueled; natural gas engines are less important. Because of the wide range of power offered, from 50 to 1500 hp, these engines are used in a wide variety of miscellaneous industrial, commercial, nonpropulsive marine, and agricultural applications where shaft power is needed and electricity is either unavailable or inappropriate (see Chapter 3.1.2).

Although precise production data are unavailable, we estimate that sales of diesel medium-power, high-speed engines for stationary applications have been in the range of 60,000 to 80,000 units/year over the past 6 years with a total value of $150 to $200 million per year (FOB plant), and annual sales for gasoline medium-power, high-speed engines have been approximately 100,000 with a total value of $50 million (FOB plant). Sales of these engines have been erratic from one year to the next, but the general trend seems to be upward. The industry expected this growth to continue at about the same range as the economy (that is, at 3 to 5 percent per year) before the recent downturn and concomitant economic uncertainties(7,8).

### 3.1.1.3 Small Engines

Five firms are classified as manufacturers of small engines and generator sets. These manufacturers are distinguished from the previously discussed manufacturers in that they produce mostly one- and two-cylinder engines of less than 50 horsepower. Although there is some overlap between the power ranges offered by the manufacturers of medium-power, high-speed engines and those of small engines, most of the engines sold by the former group are larger than 50 horsepower.

The engines produced by the five manufacturers in this group are mostly diesel and gasoline, one- and two-cylinder models. The firms also produce some four-cylinder models. All have four strokes per power cycle, and four of the firms produce air-cooled engines. Onan, Kohler, and Wills produce mainly engine-generator sets -- small, semiportable integral engines and generators that are used to provide electrical power in remote locations. Other applications for the engines include power for small pumps and blowers and off-the-road vehicles. A particularly important application for some manufacturers is refrigeration compressors for trucks and railroad cars, and hydraulic pumps for tractor-trailer dump trucks and trash compactors. Sales of small engines are in the range of $75 to $100 million per year, and employment is approximately 4000 to 6000.

#### 3.1.1.4 Very Small Engine Manufacturers

In addition to the manufacturers discussed above, there are at least 10 firms which produce one-cylinder, air-cooled gasoline engines rated at less than 20 hp. These firms sell about 12.5 million engines per year (1973) with a total product value of $400 million(9). Therefore, the unit wholesale cost of each engine is about $30. These engines are used primarily for lawn and garden equipment and chain saws, and to a lesser degree for recreational vehicles, such as snowmobiles and small all-terrain vehicles. The dollar sales of these engines constitutes 60 percent of the value of all gasoline engines (except automotive, outboard, truck, bus, tank, and aircraft) and is approximately equal to the dollar sales of all other stationary engines (estimated at $300 million to $400 million annually, as mentioned above).

### 3.1.2 Engine Users

On the basis of installed horsepower, the principal stationary applications of IC engines are the following: oil and gas pipelines, oil and gas production, general industrial (including construction), electrical power generation, and agriculture. Table 3-2(10-32) shows estimated IC engine applications and usage patterns by fuel.

## TABLE 3-2. STATIONARY ENGINE APPLICATIONS

| Category | Annual Production, units/yr | Population,$^a$ units | Average Power, hp | Load Factor | Annual Usage, hr/yr | Annual Energy Production – Yearly Sales, hp-hr/yr x $10^6$ | Annual Energy Production – Installed Units, hp-hr/yr x $10^6$ | Constant Load | Variable Load | Constant Speed | Variable Speed | Basis of Estimate$^b$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| DIESEL | | | | | | | | | | | | |
| **Oil & Gas Production** | | | | | | | | | | | | |
| Offshore drilling | 800$^c$ | 675 | 350 | 0.8 | 2000 | | 378 | | X | | X | AGA Market Study |
| Land drilling | | 3,050 | 350 | 0.8 | 2000 | | 1,708 | | X | | X | AGA Market Study |
| **Oil & Gas Transport** | 50$^c$ | 500 | 2000 | 0.8 | 6000 | | 4,800 | X | | | X | McGowin, Gas Facts |
| **Electric Generation** | 150$^c$ | 400$^c$ | 2500 | 0.8 | 2600 | | 2,160 | | X | X | | FPC, Diesel & Gas Power Costs |
| **General Industrial & Agriculture** | | | | | | | | | | | | |
| Municipal water supply | 200$^c$ | 2,100 | 120 | 0.75 | 3000 | | 567 | X | | X | | AGA Market Study |
| Marine nonpropulsive | 5,000$^d$ | 15,000 | 100 | 0.5 | 3500 | 62.5 | 1,875 | | X | | X | Current Industrial Report, Industry contacts |
| Construction, small | 5,000 | 50,000 | 50 | 0.5 | 500 | 62.5 | 625 | | X | | X | |
| Miscellaneous, large$^e$ | 1,700 | | 750 | 0.5 | 100 | 56.3 | 1,125 | | | | | |
| Construction, large | 5,000 | 50,000 | 240 | 0.5 | 500 | 300.0 | 3,000 | | X | | X | |
| Portable compressors$^f$ | 9,000 | 90,000 | 75 | 0.5 | 500 | 168.8 | 1,688 | | X | | X | |
| Welders | 8,000 | 80,000 | 55 | 0.5 | 500 | 110 | 1,000 | X | X | | X | |
| Pumps | 5,000 | 25,000 | 100 | 0.5 | 1000 | 250 | 1,250 | X | | X | | |
| Generator sets(stand by) | | | | | | | | | | | | |
| <50 kW | 7,000 | 70,000 | 75 | 0.5 | 500 | 35.0 | 350 | | X | X | | |
| 50 kW – 400 kW | 8,000 | 160,000 | 250 | 0.5 | 250 | 250 | 5,000 | X | X | X | | |
| 400 kW – 1000 kW | 1,500 | 30,000 | 750 | 0.5 | 100 | 56.3 | 1,125 | X | X | X | | |

Footnotes appear at end

### TABLE 3-2. Continued

| Category | Annual Production, units/yr | Population,[a] units | Average Power, hp | Load Factor | Annual Usage, hr/yr | Annual Energy Production – Yearly Sales, hp-hr/yr x $10^6$ | Annual Energy Production – Installed Units, hp-hr/yr x $10^6$ | Constant Load | Variable Load | Constant Speed | Variable Speed | Basis of Estimate[b] |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| DUAL FUEL | | | | | | | | | | | | |
| Oil & Gas Transport | included in Diesel | included in Diesel | | 0.8 | 6000 | | 2,228 | X | | | X | McGowin, Gas Facts |
| Electric Generation | | | | 0.8 | | | 6,000 | | X | X | | FPC, Diesel & Gas Power Costs |
| NATURAL GAS[g] | | | | | | | | | | | | |
| Agriculture | | 91,000 | 100 | 0.75 | 2500 | | 17,063 | X | | X | | AGA Market Study |
| Oil & Gas Production | | | | | | | | | | | | |
| Oil & gas well pumps | | 266,000 | 15 | 0.7 | 3500 | | 9,776 | X | | X | | AGA Market Study |
| Secondary recovery | | 5,600 | 200 | 0.8 | 6000 | | 5,376 | X | | X | | AGA Market Study |
| Well drilling | | 3,050 | 350 | 0.8 | 2000 | | 1,708 | X | | X | | AGA Market Study |
| Plant processing | | 4,000 | 750 | 0.8 | 8000 | | 19,200 | X | | X | | McGowin |
| Oil & Gas Industry | | | | | | | | | | | | |
| Utility compressors[h] | | 4,500 | 2000 | 0.9 | 6000 | | 51,800 | X | | X | | Southwest Research Institute[i] |
| | | 4,000 | 750 | 0.8 | 6000 | | 14,400 | X | | X | | Southwest Research Institute[i] |
| Electric Generation | | | | | | | | | | | | |
| Private/public utility | | | | 0.8 | | | 166.5 | | X | X | | FPC, Diesel & Gas Power Costs |
| Commercial-institutional | | 450 | 200 | 0.45 | 4000 | | 162 | | X | X | | AGA Market Study |
| Stand-by | | 2,000 | 100 | 0.9 | 50 | | 9 | X | | X | | AGA Market Study |
| Industrial on-site | | 1,500 | 300 | 0.6 | 4000 | | 1,080 | | X | X | | AGA Market Study |

Footnotes appear at end

## TABLE 3-2. Continued

| Category | Annual Production, units/yr | Population,[a] units | Average Power, hp | Load Factor | Annual Usage, hr/yr | Annual Energy Production – Yearly Sales, hp-hr/yr x $10^6$ | Annual Energy Production – Installed Units, hp-hr/yr x $10^6$ | Constant Load | Variable Load | Constant Speed | Variable Speed | Basis of Estimate[b] |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **General Industrial** | | | | | | | | | | | | |
| Industrial shaft power | | 2,900 | 200 | 0.75 | 5000 | | 2,175 | | X | | X | AGA Market Study |
| Plant air | | 750 | 100 | 0.5 | 4000 | | 150.4 | | X | | X | AGA Market Study |
| Air-conditioning | | 3,760 | 80 | 0.4 | 2000 | | 240.6 | | X | | X | AGA Market Study |
| Commerical shaft power | | 600 | 2000 | 0.6 | 1000 | | 720 | | X | | X | AGA Market Study |
| Municipal | | | | | | | | | | | | |
| water supply | | 2,100 | 120 | 0.75 | 3000 | | 567 | X | | X | | AGA Market Study |
| waste treatment | | 1,740 | 400 | 0.45 | 4000 | | 1,566 | X | | X | | AGA Market Study |
| GASOLINE | | | | | | | | | | | | |
| **Agriculture** | | | | | | | | | | | | |
| Misc. machinery[j] | 20,000 | 400,000 | 30 | 0.5 | 200 | 60 | 1,200 | X | X | X | X | Current Industrial Reports, Industry contacts |
| Irrigation | 5,000 | 10,000 | 100 | 0.75 | 2000 | 750 | 1,500 | X | X | X | | |
| **General Industrial** | | | | | | | | | | | | |
| Generator sets >5 kW | 35,000 | 350,000 | 55 | 0.5 | 400 | 715 | 7,150 | | X | X | | |
| Compressors | 7,000 | 70,000 | 55 | 0.5 | 400 | | | | X | | X | |
| Welders | 18,000 | 180,000 | 55 | 0.5 | 400 | | | | | | | |
| Miscellaneous | 5,000 | 50,000 | 55 | 0.5 | 400 | | | X | X | X | X | |
| Construction | 5,000 | 40,000 | 150 | 0.5 | 500 | 188 | 1,500 | | X | | X | |
| Small (<15 hp) | 12,600,000[k] | 63,000,000 | 4.2 | 0.5 | 50 | 1100 | 6,615 | X | X | X | X | Current Industrial Reports |

See footnotes on following page.

## TABLE 3-2. Concluded

Footnotes

[a] Annual production multiplied by life in years (based on estimated service life of 5000 hours for diesel engines, 4000 hours for gasoline engines, or as noted) to compute population

[b] See References 10 through 32 for complete titles of these sources.

[c] Approximated, based on estimated population and annual usage

[d] 7500-hour service life assumed

[e] Applications include pumps, snow blowers, aircraft turbine starters, etc.

[f] Excludes mobile refrigeration units

[g] Population estimates come from the AGA market study. Annual production is not estimated for this category since production has been changing rapidly, decreasing continuously since 1966 (see Figure 8-3) and, therefore, an annual estimate of production would be misleading.

[h] Includes transport, distribution, gathering, and storage

[i] Reference 24

[j] Pull combines, balers, sprayers, dusters, etc.

[k] Estimated service life of 5 years

Pipeline installations are concentrated in the oil and gas producing states on the Gulf Coast and in the Midwest as shown in Figure 3-5(a)(33). Electric generation includes base load generation, principally for municipalities in the plains and Midwest, emergency standby for vital public services and large buildings in urban areas, and remote generation (for mines and homes) in rural areas. Agriculture applications are primarily irrigation pumping, concentrated in those areas with irrigated farm lands, as shown in Figure 3-5(b)(34). Additional agricultural applications include frost control, harvesting (auxiliary engines), and some remote electric generation. Construction applications include portable compressors, welders, pumps, electric generation, and material handling equipment.

For applications other than electric power generation, IC engines often compete with electric motors. Engines involve a higher initial investment but have an advantage in locations where gas and fuel oil are less expensive than electricity, such as remote or temporary construction sites. Thus, engines tend to be concentrated in areas where electric power generation and transmission costs are high, as in the mountain and prairie states, or where fuel costs are low, as in the gas-producing areas along the Gulf Coast.

## 3.2 PROCESSING AND THEIR EMISSIONS

### 3.2.1 Fundamental Description of IC Engines[2/]

Reciprocating internal combustion engines produce shaft power by confining a combustible mixture in a small volume between the head of a piston and its surrounding cylinder, causing this mixture to burn, and

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[2/] Much of the material in this section is drawn from References 35 and 36. These books describe the operation of IC engines in considerable detail.

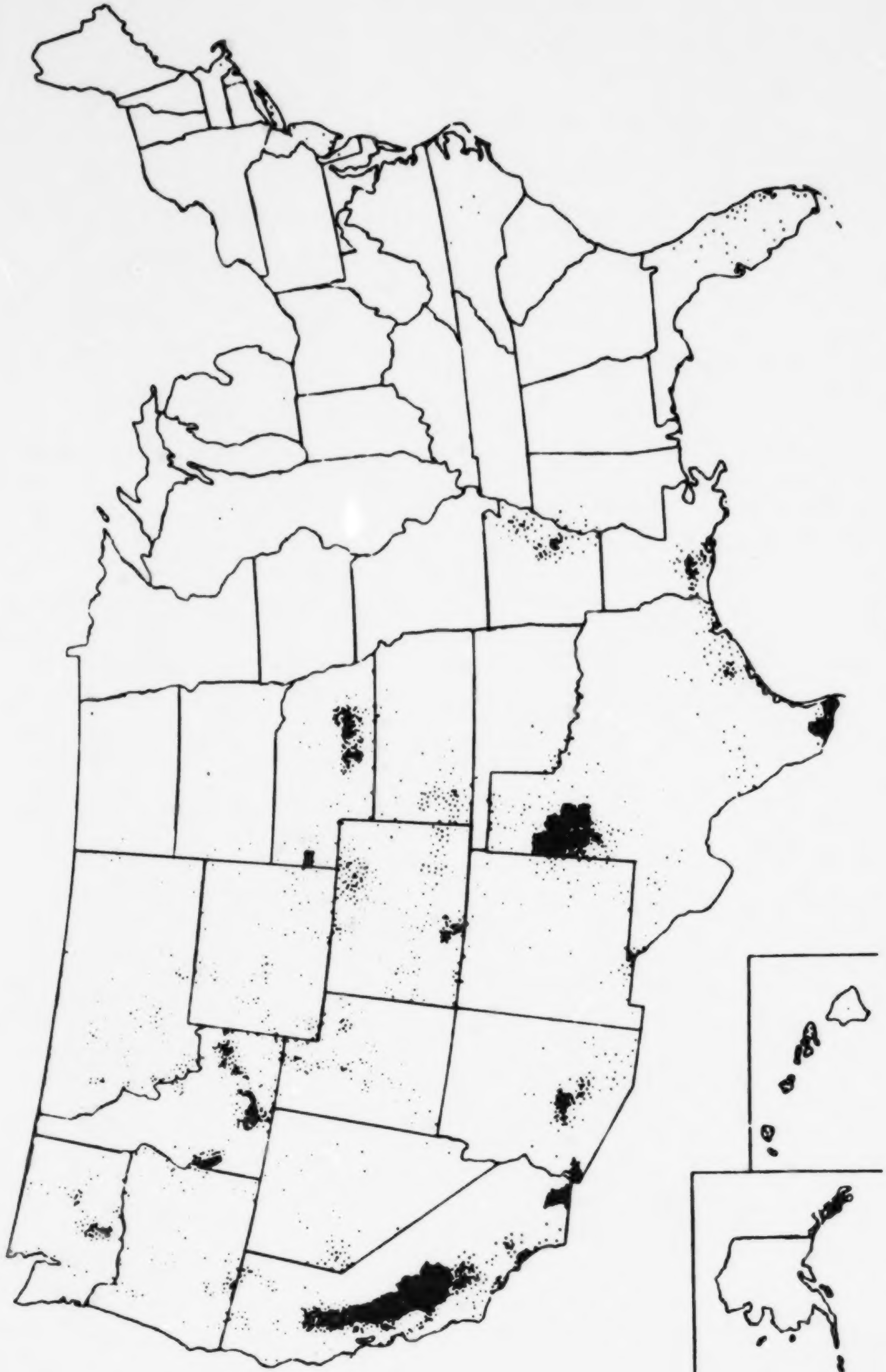

Figure 3-5(a). Potential users of stationary reciprocating engines: irrigated farms.(33)

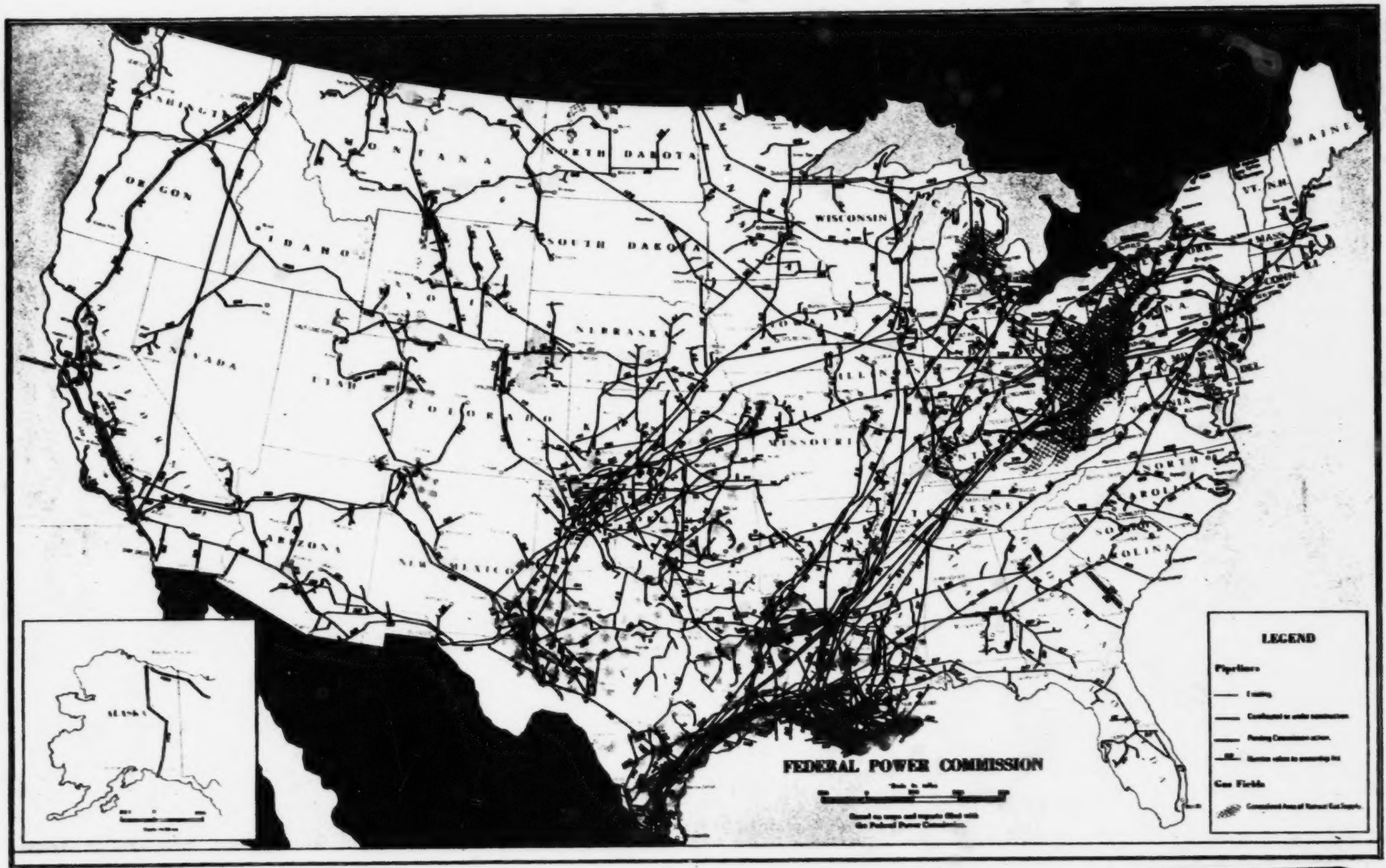

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Figure 3-5(b). Major natural gas pipelines(34).

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allowing the resulting high-pressure products of combustion gases to push the piston. Power is converted from linear to rotary form by means of a crankshaft.

There are two methods of igniting the fuel and air mixture. In compression ignition (CI) engines, air is compression heated in the cylinder, and diesel fuel is then injected into the hot air. Ignition is spontaneous. A different approach, however, is used for gasoline or gas-fueled engines where combustion is initiated by the spark of an electrical discharge. Therefore, these units are spark ignition (SI) engines. Usually the fuel is mixed with the air in a carburetor (for gasoline) or at the intake valve (for natural gas), but occasionally the fuel is injected into the compressed air in the cylinder. Although all diesel-fueled engines are compression ignited and all gasoline and gas-fueled engines are spark ignited, gas can be used in a compression ignition engine if a small amount of diesel fuel is injected into the compressed gas-air mixture to initiate burning. Such "dual fuel" engines are usually designed to burn any mixture ratio of gas and diesel oil, from 6- to 100-percent oil (based on heating value).

CI engines can operate at a higher compression ratio (ratio of cylinder volume when the piston is at the bottom of its stroke to volume when it is at the top) than SI engines because fuel is not present during compression and hence there is no danger of premature auto-ignition. Since the thermal efficiency of an engine increases with increasing pressure ratio (and pressure ratio varies directly with compression ratio), CI engines are more efficient than SI ones. This increased efficiency is gained at the expense of poorer acceleration and a heavier structure to withstand the higher pressures.

In addition, engines may be described by the number of strokes per cycle (two or four) and the method of introducing air and fuel into the cylinder.

In the four-stroke cycle, the sequence of events may be summarized as follows (see Figure 3-6)(37):

1. Intake stroke -- suction of the air or air and fuel mixture into the cylinder by the downward motion of the piston
2. Compression stroke -- compression of the air or air and fuel mixture, thereby raising its temperature
3. Ignition and power (expansion) stroke -- combustion and consequent downward movement of the piston with energy transfer to the crankshaft
4. Exhaust stroke -- expulsion of the exhaust gases from the cylinder by the upward movement of the piston

This description applies to a naturally aspirated engine, which utilizes the vacuum created behind the moving piston to suck in the fresh air charge. However, many engines now pressurize the air cylinder. This may be done with either a turbocharger or a supercharger. The turbocharger is powered by a turbine that is driven by the energy in the relatively hot exhaust gases, while a supercharger is driven off the engine crankshaft. Air pressurization is used to increase the power density, or power output per unit weight (or volume) of the engine. Since the density of air increases with pressure, the mass of air that can be introduced into the cylinder increases with pressure. Furthermore, since the air-to-fuel ratio at maximum power is fixed by combustion requirements, more fuel can be introduced into the cylinder with high pressure air than with atmospheric pressure air; hence, more power can be obtained from a given cylinder configuration. As the air pressure is

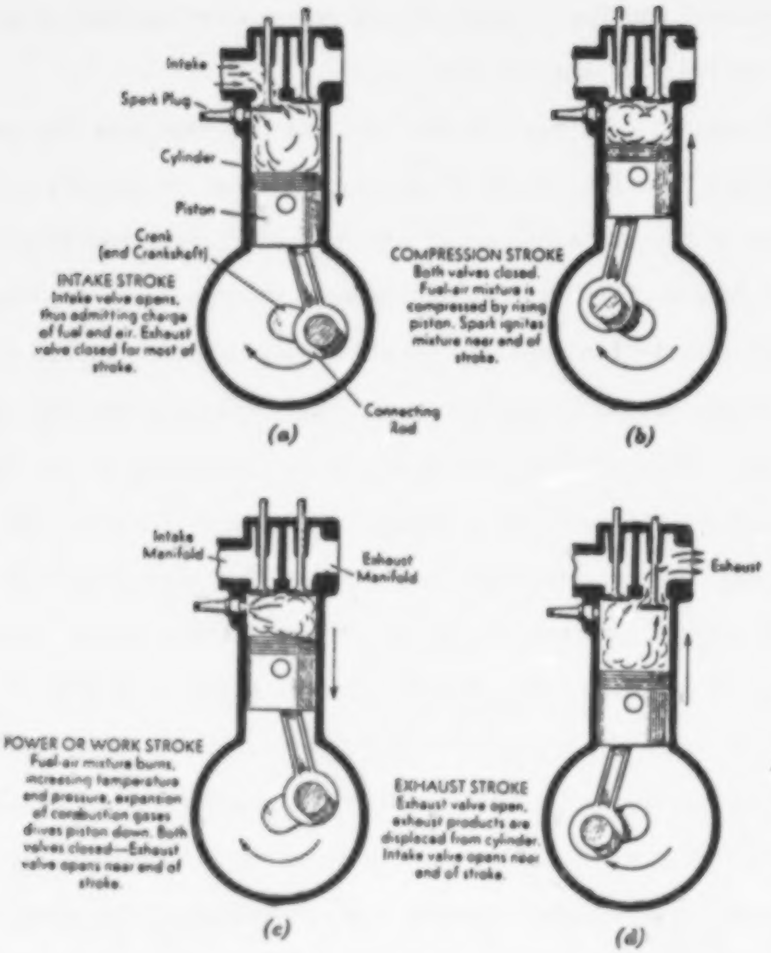

Intake
Spark Plug
Cylinder
Piston
Crank (and Crankshaft)
INTAKE STROKE
Intake valve opens, thus admitting charge of fuel and air. Exhaust valve closed for most of stroke.
Connecting Rod
(a)
COMPRESSION STROKE
Both valves closed. Fuel-air mixture is compressed by rising piston. Spark ignites mixture near end of stroke.
(b)
Intake Manifold
Exhaust Manifold
POWER OR WORK STROKE
Fuel-air mixture burns, increasing temperature and pressure, expansion of combustion gases drives piston down. Both valves closed—Exhaust valve opens near end of stroke.
(c)
Exhaust
EXHAUST STROKE
Exhaust valve open, exhaust products are displaced from cylinder. Intake valve opens near end of stroke.
(d)

Figure 3-6. The four-stroke, spark-ignition (SI) cycle. Four strokes of 180° of crankshaft rotation each, or 720° of crankshaft rotation per cycle(37).

increased, its temperature is also raised. Therefore, the pressurized air is often cooled before entering the cylinder to further increase power. This process is called intercooling or aftercooling. In fact, all high power turbocharged natural gas-fueled engines are intercooled to prevent premature auto-ignition of the fuel and air mixture(38).

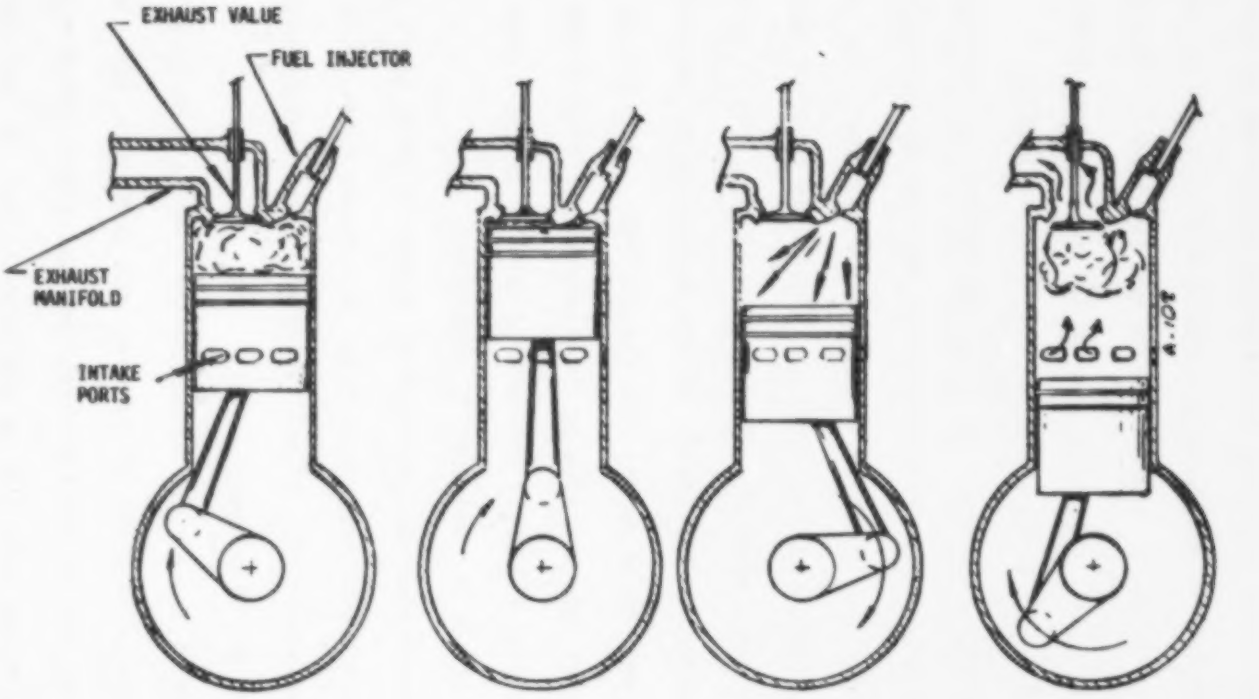
In a CI engine, fuel is injected into the cylinder near the end of the compression stroke; whereas, in an SI engine, the fuel is usually added to the air, downstream of the turbocharger if any is used, before the mixture enters the cylinder. In some SI engines (particularly large natural gas-fueled ones), the fuel is injected into the intake manifold just ahead of the valves. For CI engines or SI engines using fuel injection instead of carburetors, the fuel or fuel-air mixture can be introduced by "direct injection" into the cylinder head and communicates with the principal combustion volume. Direct injection units are also called open chamber engines because combustion takes place in the open volume between the top of the piston and the cylinder. This design contrasts with indirect injection engines, where the combustion begins in a fuel-rich (oxygen deficient) atmosphere in the smaller antechamber and then expands into the cooler, excess air region of the main chamber. These latter engines are also called divided or precombustion chamber systems (swirl chamber if the small antechamber is specially designed to promote swirl -- see Section 4.4.8). Prechamber designs can be used in carbureted engines as well, but they have not yet been applied to engines built exclusively for stationary applications.

There are three major advantages of divided chamber engines: less sensitive to fuel variations; lower stresses on the mechanical parts, such as connecting rods, crankshaft, etc., because of a lower pressure rise rate during combustion; and lower $NO_x$ production. All three of these benefits are a result of incomplete primary combustion with an oxygen deficiency. Therefore, by comparison with open chamber designs, combustion is less sensitive to fuel droplet spray patterns, the complete combustion process takes longer, and less oxygen is available to combine with nitrogen in the initial high-temperature reaction. These benefits are attained at the expense of up to 30 percent greater heat rejection (due to the surface area added by the antechamber) and consequent 5- to 8-percent fuel penalty(39,40).

In a two-stroke design, the power cycle is completed in one revolution of the crankshaft as compared to two revolutions for the four-stroke cycle (see Figure 3-7). As the piston moves to the top of the cylinder, air, or an air and fuel mixture, is compressed for ignition. Following ignition and combustion the piston delivers power as it moves down. Eventually it uncovers the exhaust ports (or exhaust valves open). As the piston begins the next cycle, exhaust gas continues to be purged from the cylinder, partially by the upward motion of the piston and partially by the scavenging action of the incoming fresh air. Finally, all ports are covered (and/or valves closed), and the fresh charge of air or air and fuel is compressed.

Air charging in two-stroke designs is often accomplished by means of a low-pressure blower, which also aids in purging the exhaust gases; such systems are called blower-scavenged. Naturally aspirated and turbocharged (or supercharged) systems are also common.

The main advantage of two-stroke engines is their horsepower-to-weight ratio as compared to four-stroke prime movers when both operate at the same speed. This is, of course, due to the fact that the two-stroke design has twice as many power strokes per unit time as the four-stroke. In addition, if ports are used instead of valves, the mechanical design of the engine is



COMPRESSION
A. INTAKE PORTS COVERED
B. EXHAUST VALVE CLOSES

COMBUSTION
A. FUEL ENTERS CYLINDER BY INJECTION
B. COMBUSTION BY AUTO-IGNITION

POWER STROKE
A . PISTON MOVES DOWN
B. POWER DELIVERED TO CRANKSHAFT

SCAVENGING
A. AIR BLOWN INTO CYLINDER
B. EXHAUST GASES PURGED

Figure 3-7. Cylinder events for a two-stroke blower-scavenged IC engine.

simplified. However, combustion cannot be controlled as well, and excess air is needed to purge the cylinder. Therefore, these engines tend to be slightly less efficient, and uncontrolled models tend to emit slightly more pollutants (primarily unburned hydrocarbons), than do their four-stroke counterparts(41).

### 3.2.2 Emission Sources and Types in IC Engines

Most of the pollutants from IC engines are emitted through the exhaust. However, some hydrocarbons escape from the crankcase as a result of blowby (gases which are vented from the oil pan after they have escaped from the cylinder past the piston rings) and from the fuel tank and carburetor because of evaporation.[3/] Nearly all of the hydrocarbons from diesel (compression ignition) engines enter the atmosphere from the exhaust(43). Crankcase blowby is minor (0.18 g/hp-hr HC + $NO_x$, 0.0063 g/hp-hr CO) because hydrocarbons are not present during compression of the charge(44,45). Evaporative losses are insignificant in diesel engines due to the low volatility of diesel fuels(46). In general, evaporative losses are also negligible in engines using gaseous fuels because these engines receive their fuel continously from a pipe rather than via a fuel storage tank and fuel pump(47). In gasoline-fueled engines, however, 20 to 25 percent of the total hydrocarbon emissions from uncontrolled engines come from crankcase blowby and another 10 to 15 percent from evaporation of the fuel in the storage tank and the carburetor (divided approximately equally between the two)(48).

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[3/] Until recently it was thought that HC evaporative controls on automobile fuel tanks and carburetors reduced emissions from these sources to about 5 percent of the uncontrolled levels. However, further, more comprehensive testing has shown that controlled vehicles actually allow HC to be evaporated at approximately 70 percent of the rate from uncontrolled cars (42).

However, crankcase blowby emissions can be virtually eliminated through the simple expedient of the positive crankcase ventilation (PCV) valve.

The primary pollutants from internal combustion engines are oxides of nitrogen (NOx), hydrocarbons and other organic compounds (HC), carbon monoxide (CO), and particulates, which include both visible (smoke) and nonvisible emissions. $NO_x$ formation is directly related to high pressures and temperatures during the combustion process and to the nitrogen content of the fuel. The other pollutants, HC, CO, and smoke, are primarily the result of incomplete combustion. Specific descriptions of these formation mechanisms in SI and CI engines are discussed in Chapter 4. Ash and metallic additives in the fuel also contribute to the particulate content of the exhaust, but usually do not cause the exhaust to be visible. Although the odor of the exhaust (primarily a problem with diesels) is believed to be the result of some organic compounds in this exhaust (particularly aldehydes), a precise relation has not yet been established between exhaust and odor(49-52). Considerable research has led to the belief that aldehydes may be one cause of odor in diesel and gasoline exhaust. Aromatic hydrocarbons and other unburned elements of fuel, as well as sulfur species, may also be a source of odor(53).

Oxides of sulfur, lead, and other metals also appear in the exhaust from IC engines. The sulfur compounds, mainly $SO_2$, are directly related to the sulfur content of the fuel. Most of the metals are fuel additives and generally leave the engine as particulates. Lead is commonly added to gasoline lines as an antiknock agent. However, laed-free gasoline is now available nationwide, and leaded gasoline may not be marketed in quantity in the future. Although various metals, particularly barium, are sometimes added to diesel fuel as a smoke suppressant, their use is not recommended by most engine manufacturers because of possible adverse effects on the engines and on the environment(54,55).

### 3.2.2.1 Emission Characteristics -- Spark Ignition Engines

Typical uncontrolled emission rates (23-mode composite)(56) for four-stroke naturally aspirated gasoline engines in g/brake hp-hr are: 2.4 to 13.1 for HC; 30.4 to 89.8 for CO; and 7.9 to 13.9 for $NO_x$(57). Similarly, brake specific emission rates from all types of spark-ignited gas engines (two- and four-stroke, naturally aspirated and turbocharged engines) are: 0.4 to 1.9 for HC; 0.2 to 0.6 for CO; and 15.0 to 32.0 for $NO_x$(58).

Hydrocarbon emissions from these engines may be partially burned or completely unburned. Three general mechanisms are believed to be the cause of these pollutants(59). The first, wall quenching, arises when fuel droplets impinge on the cylinder walls and are cooled below the ignition temperature. Whether the hydrocarbons remain unburned or are partially oxidized depends upon the wall temperature, the droplet temperature at impingement, and the combustion intensity nearby. The second source of HC emission, incomplete combustion, can be caused by one, or more, of the following:

a. Change in charge homogeneity

b. Incorrect air-to-fuel ratio (too rich or too lean)

c. Low temperature of the incoming air-fuel mixture

d. Failure of the engine to purge all the products of combustion

e. Defective ignition system

Scavenging, or the displacement of exhaust gases in the cylinder by a fresh charge of air or air and fuel, is the third mechanism resulting in HC emissions. It is especially a factor in carbureted two-cycle engines, where part of the intake charge is swept out with the exhaust completely escaping the combustion process.

Carbon monoxide emissions are also related to incomplete combustion. CO is formed during combustion and then normally combines with oxygen atoms (dissociated oxygen molecules) to form $CO_2$, the final product. This last reaction can only proceed at a high temperature (above 900°K)(60). Therefore, if the overall mixture is rich or if sufficient oxygen is not available during and after combustion, before the products of combustion cool below 900°K, CO will be formed. Thus, both CO and HC are due to improper air-fuel mixing and improper introduction into the chamber. Their generation is also a measure of the inefficiency of the combustion process.

$NO_x$ formation, on the other hand, increases with increasing combustion efficiency. Its generation is governed by the pressure, temperature and oxygen concentration during combustion. Nitrous oxide (NO) is generated in the cylinder when both the $N_2$ and the $O_2$ molecules dissociate into free atoms at the high temperature encountered during combustion, and then recombine to form NO. The reaction rate toward NO increases exponentially with temperature. Maximum temperatures occur when the air-to-fuel ratio is just above stoichiometric. $NO_2$ and other $NO_x$ compounds form later in the presence of additional oxygen. $NO_x$ emissions are particularly high from gas engines, which burn fuel with an excess of oxygen. In any engine, as the air-to-fuel ratio decreases from stoichiometric, $NO_x$ formation decreases because of a lack of excess oxygen. As the air-to-fuel ratio increases from stoichiometric, $NO_x$ formation first increases due to the presence of more oxygen. Further increases in air flowrate cause a decline in peak combustion temperature and a consequent decrease in $NO_x$ formation (Figure 3-8)(61).

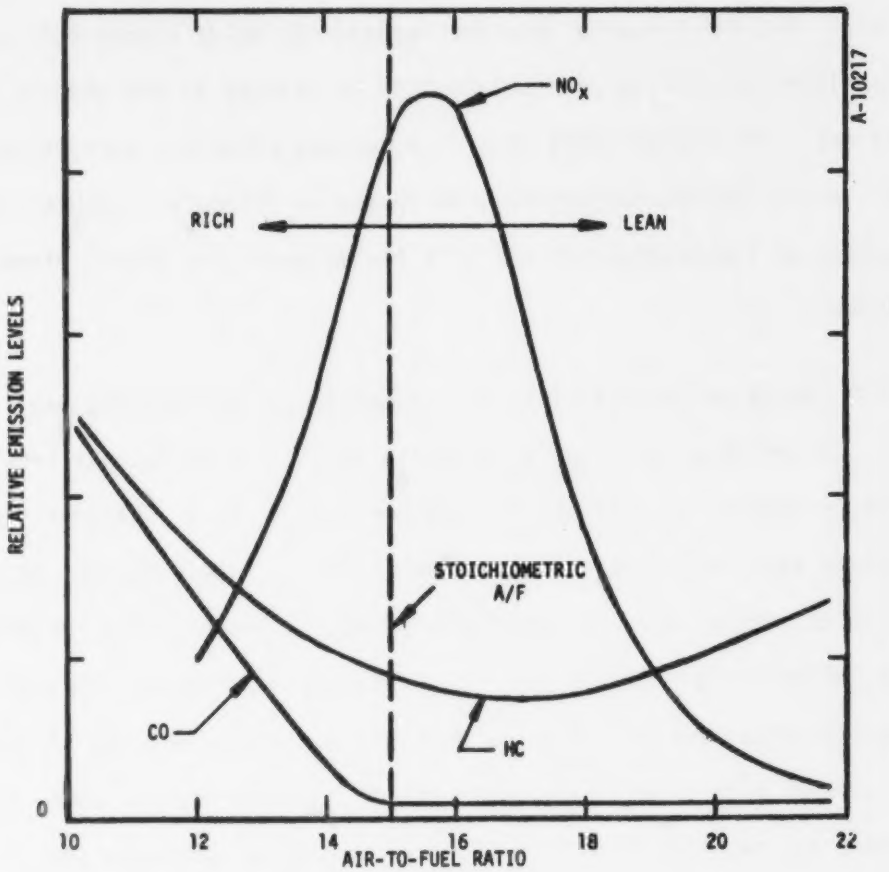

Figure 3-8. Effect of A/F ratio on emissions from a gasoline engine(61).

Visible emissions (smoke) are generally insignificant in gasoline and gas engines. Particulates, however, may be present, especially as a result of rich fuel mixtures or impurities in the fuel. On a mass basis, the majority of particulate emissions from gasoline engine exhausts are now the result of lead in the fuel(62). When smoke is present, particulate levels are high, but the converse does not necessarily hold; significant quantities of non-light-scattering particulates can be emitted in the absence of visible emissions. The bluish smoke that is sometimes observed, particularly in older, poorly maintained engines, is caused by incomplete combustion of crankcase or lubricating oil which is forced past worn piston rings into the cylinder.

### 3.2.2.2 Emission Characteristics -- Compression Ignition Engines

As mentioned previously, essentially all the pollutants from diesel or dual-fuel engines are emitted through the exhaust (i.e., evaporative and crankcase ventilation emissions are negligible). Again HC, CO, $NO_x$ and particulate matter are the pollutants of major concern. The formation of these pollutants is complex and not completely understood. Typical emission rates (specific) for all types of uncontrolled diesel engines (including four-stroke, either naturally aspirated or turbocharged and open- or divided-chamber, and two-stroke, either blower-scavenged or turbocharged) in g/brake hp-hr are: 0.1 to 2.9 for HC; 0.3 to 14.6 for CO; and 2.1 to 17.1 for $NO_x$(63). Unburned and partially burned hydrocarbons are formed in CI engines by many of the same mechanisms that contribute to their generation in SI engines. There are, however, two major differences: the fuel injection system and the combustion chamber design. Fuel is introduced into the cylinder of a CI engine by a fuel injector, independently of the air, whereas a fuel and air mixture generally is allowed to flow into the cylinder of an SI engine through the intake valve or port. Therefore, the fuel-air mixture in a CI engine is heterogeneous by design. However, the fuel distribution within the cylinder can be controlled somewhat by the design of the injector, and therefore, both wall quenching and large-scale inhomogeneities can be minimized(64,65). Hence, both the fuel injection system and the combustion chamber design are major factors which determine the quantity of HC emitted(66,67).

Since the carbon monoxide results from incomplete combustion in the absence of sufficient oxygen, and since CI engines are designed to operate lean (excess air), CO emissions are usually low.

Conditions within the cylinder of a CI engine during combustion favor the production of $NO_x$, because it is formed at high temperatures and pressures in the presence of oxygen. However, high temperatures and pressures lead to high thermal efficiency (low fuel consumption) and high air-to-fuel ratios lead to low smoke, HC, and CO generation(68,69). Therefore, design compromises are usually required when an attempt is made to reduce or limit $NO_x$ emissions.

Smoke occurs in three forms: white (cold) smoke, blue, and black (hot smoke)(70). The white smoke consists primarily of unburned liquid fuel or lubricating oil and occurs at low load or idle conditions. Blue smoke is caused by lubrication oil that leaks into the combustion chamber. Black smoke consists of carbon particles that result from incomplete combustion at high temperatures. Incomplete combustion is possible under high loads (low air-to-fuel ratio) and poor mixing of the fuel-air mixture. Emission rates of white and blue smoke are usually not measured by manufacturers. They generally report black smoke emission rates of less than 10-percent opacity

(1.18 milligrams per cubic foot for a 16-inch diameter stack) for a well maintained engine(71-73).

### 3.2.2.3 Effects of Variables on Emissions

Many variables affect emissions from IC engines. Table 3-3(74) lists these variables, divided into three categories. Those listed as "Design" are features that are normally determined by the manufacturer. The "Operation Adjustment" category represents variables that can be altered by the user. Since these variables can be adjusted to reduce emissions, they are discussed in more detail in Chapter 4, Emission Control Techniques. The last column lists those variations in ambient air conditions which affect emissions.

### 3.2.2.3.1 Size

Engine size has a definite effect on emissions. Large-bore, low- to medium-speed engines, independent of the fuel type (diesel, dual fuel, natural gas), have certain common emission characteristics. These large-bore engines are designed for low fuel consumption (i.e., high thermal efficiency). Furthermore, their low rotational speed maintains the products of combustion near peak temperature for relatively long periods of time. These slow speeds allow the designer to "tailor" his fuel injection schedule for optimum results, and the large combustion volumes also give him more aerodynamic design freedom than is available to the manufacturer of small- or medium-bore engines(75). Furthermore, these large engines are designed to operate under steady conditions and almost always at more than 50 percent of rated power; therefore, they do not have to contend with acceleration/deceleration or low power requirements(76,77).

TABLE 3-3. FACTORS THAT AFFECT EMISSIONS FROM RECIPROCATING ENGINES(74)

| Design | Operation Adjustment | Ambient Conditions |
|---|---|---|
| Surface to volume ratio | Air-to-fuel ratio | Pressure (altitude) |
| Bore and stroke | Torque (mean effective pressure) | Temperature |
| Valve overlap | Speed | Humidity |
| Displacement/cylinder | Spark timing | |
| Strokes/power cycle | Fuel injection timing | |
| Chamber design | Fuel properties | |
| Compression ratio | Lubrication and maintenance | |
| Air charging: | | |
| Naturally aspirated | | |
| Blower scavenged | | |
| Turbocharged | | |
| Fuel charging: | | |
| Direct injection | | |
| Indirect injection | | |
| Carbureted | | |
| Engine cooling | | |

#### 3.2.2.3.2 Load and Air-to-Fuel Ratio

Load and air-to-fuel (A/F) ratio variations have a significant effect on $NO_x$, CO, HC, and particulate emissions. In general, diesel compression ignition engines exhibit decreasing brake specific emissions of $NO_x$ with increasing load (i.e., the lower A/F ratio and higher power output overcome the potential for increased $NO_x$ production due to higher temperatures)(78). CO emissions first decrease with increasing load (equivalent to increasing temperature) and then increase dramatically as maximum load is approached. Incomplete combustion with the decreasing A/F ratio (increasing load) is largely responsible for the final trend reversal. Brake specific HC emissions, likewise, decrease with increasing load as a result of increasing temperature(79). Small (less than 15 hp) SI engines (gasoline) exhibit relatively high HC and CO emissions, particularly at low load operation(80).

Figure 3-8 illustrated the effect of A/F ratio upon emissions from spark ignition engines. $NO_x$ and CO emissions are very definitely related to A/F ratios. These trends hold throughout the load range for an SI engine. Note that an engine whose A/F ratio is adjusted to yield lowest HC and CO emissions has a maximum $NO_x$ emission. An uncontrolled engine that is adjusted to near-stoichiometric conditions will emit relatively large quantities of $NO_x$, whereas one that is run either lean or rich will produce significant quantities of HC and CO (assuming other modifications or control technologies are not applied to control HC and CO).

#### 3.2.2.3.3 Climate and Location

All the effects of variations in climate and geographic location on emissions are reflected by changes in the pressure, temperature, and humidity of the ambient air. Pressure and temperature variations alter the density of the inlet air charge and, hence, change the air-to-fuel ratio. (The consequences of changing A/F ratio have just been discussed.) Secondarily, they change the peak combustion pressure and temperature in proportion to their variations in the ambient air. Since $NO_x$ production is a strong function of temperature, variations of the inlet air temperature have a noticeable effect on $NO_x$ emissions. In general, humidity has little effect upon HC and CO emissions(81,82). $NO_x$ emissions, however, are significantly affected by humidity variations because the water absorbs some of the heat generated during combustion, thereby reducing the peak combustion temperature. Although the qualitative effects of varying ambient conditions of emissions are known, a reliable quantitative relationship exists only for the effects of humidity on $NO_x$ production from vehicular gasoline and diesel engines(83,84). Since these relationships are based on experiments, and cannot be predicted analytically, they are applicable only to stationary engines which are similar to vehicular models. Similar relationships have not yet been developed for high-power, low- and medium-speed engines. Since the combustion characteristics of these large engines are different than those of the vehicular size units, there is no a priori justification for applying the above-mentioned relationship to the larger engines.

#### 3.2.2.3.4 Fuel Effects

Significant variations in the emissions of reactive HC may be due to the fuels used. As mentioned previously, HC emissions from SI gasoline engines originate from the fuel system through evaporation, from the exhaust due to combustion, and from crankcase blowby. In general, fuel evaporation is proportional to fuel volatility(85). Effective systems are now used to control evaporative emissions and crankcase blowby on automobiles. Studies have shown that exhaust HC emissions from unleaded fuels have a higher photochemical reactivity than do those from base leaded fuels, assuming no catalytic converters are used(86).

Diesel fuels used in CI engines may display some variation in density and viscosity which, in turn, affects the A/F ratio and fuel spray pattern in the cylinder. If only a single fuel is used, the engine is "tuned" for it, but is is unlikely that engines would be "retuned" for each fuel if fuel types were frequently changed.

Particulate and smoke generation tends to increase with increasing Cetane number. The relationship between smoke and fuel volatility is unclear because it is difficult to separate the effects due to volatility changes from those due to Cetane number(87).

### 3.2.3 State and Local Regulations

Most states and localities restrict visible emissions from stationary sources, including reciprocating internal combustion engines, to less than 20-percent opacity and many are now requiring that plumes from new installations have less than 10-percent capacity(88). These limits usually do not present a problem to the owner who operates and maintains his engine correctly. However, since many $NO_x$ control systems tend to increase smoke, engines may not be able to meet these opacity limits if very stringent $NO_x$ standards are proposed.

The only known restrictions on gaseous emissions from reciprocating IC engines (i.e., excluding gas turbines, which have been specifically regulated by some localities) are in the Los Angeles area. Several control districts in this basin limit $NO_x$ emissions (expressed as $NO_2$) to 140 lb/hr (63.5 kg/hr)(89-91). Typical uncontrolled engines emit 10 to 20 g/hp-hr (see

Chapter 5). At 15 g/hp-hr these $NO_x$ limits would prevent a user from buying an uncontrolled engine that is larger than about 4250 hp (3200 kW). As is also shown in Chapter 5, emissions from a few large-bore engines can be brought down to levels as low as 8 g/hp-hr by control techniques that can be applied in the near term. Such engines could be as large as 7950 hp (6000 kW) and still be allowed to operate in the Los Angeles basin.

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# CHAPTER 4

# EMISSION CONTROL TECHNIQUES

Stationary reciprocating IC engines emit primarily $NO_x$, CO, HC, and particulates (see Section 3.2.2). All of these emissions are criteria pollutants; that is, National Ambient Air Quality Standards (or maximum allowable ambient concentrations) have been established for these pollutants under Sections 108 and 109 of the Clean Air Act. As indicated in Section 9.1, EPA has placed a high priority on the control of $NO_x$ emissions from stationary sources since these emissions are projected to increase about 25 percent by 1985 despite the application of best available control technology to new sources. Moreover, $NO_x$ is the most significant pollutant emitted from stationary IC engines, accounting for more than 16 percent of the total $NO_x$ emitted from stationary sources. In addition, as will be shown in Section 9.1, approximately 75 percent of the $NO_x$ emissions from installed stationary reciprocating IC engines (1974) was contributed by units larger than 500 hp. This chapter, therefore, emphasizes the control of $NO_x$, and control techniques are discussed which have been applied, or can potentially be applied, to large, stationary IC engines.

This chapter is divided into six sections. The formation of the pollutants emitted by IC engines is briefly discussed in Section 4.1 to aid in understanding emissions control. Section 4.2 discusses the effect of ambient conditions (humidity, temperature, and pressure), and measurement practices upon reported emission levels. Specifically, Section 4.2.1 describes how ambient correction factors will be used to adjust reported $NO_x$ data to standard conditions, and Section 4.2.2 discusses and compares the measurement practices of the manufacturers who reported emission data. The uncertainty of each stationary IC engine manufacturer's measurement practice is estimated relative to EPA's proposed approach for 1979 and later model heavy-duty diesel and gasoline engines. These uncertainties will be used to place boundaries on estimates of average emissions from engines of the same fuel type, but different manufacturer.

Section 4.3 presents uncontrolled (baseline) emission levels for large stationary IC engines. In addition to these data, sales-weighted averages of uncontrolled $NO_x$ levels (by horsepower, excluding standby applications) will be presented.

Section 4.4 discusses specific control systems to reduce $NO_x$ emissions. The discussion considers how each control system works: its effectiveness, resulting fuel penalties, effects on other emissions (HC, CO, smoke), technical limitations to its application, and cost implications (fuel, hardware, and maintenance). The emission data are corrected to standard ambient conditions where possible (see Section 4.2.1).

Finally, control systems for HC and CO emissions are discussed in Section 4.5, and techniques to reduce visible emissions are reviewed in Section 4.6. As stated in Section 4.5, there has been little effort by manufacturers of large-bore engines to reduce emission rates of HC and CO from their engines, primarily because there are currently no regulations affecting these emissions and they are already quite low. Moreover, the application of $NO_x$ control systems has, in general, only a small effect on HC and CO levels (see Section 4.4.12). Furthermore, several manufacturers have achieved reductions in HC and CO levels as a consequence of their efforts to reduce visible emissions and improve the fuel economy of their diesel engines. These practices are discussed in both Sections 4.5 and 4.6.

## 4.1 POLLUTANT FORMATION

The combustion of fuel and air in the cylinder of a reciprocating engine results in the formation of a number of chemical compounds which can "cause or contribute to the endangerment of public health or welfare" when emitted to the atmosphere. The ones of most concern fall into the general categories of nitrogen oxides ($NO_x$), carbon monoxide (CO), sulfur oxides ($SO_x$), various hydrocarbons and organic compounds (HC), and particulates. Additionally, smoke and odorous fumes may be produced, chiefly in compression ignition engines. The following paragraphs discuss the formation of these pollutants in stationary reciprocating IC engines.

### 4.1.1 Nitrogen Oxides ($NO_x$)

Virtually all $NO_x$ emissions originate as nitric oxide (NO) resulting from oxidation of nitrogen during the combustion process. Both $N_2$ and $O_2$ dissociate into atomic N and O, respectively, at the pressures and temperatures encountered in the cylinder. These atoms combine to form NO, part of which is then further oxidized in the engine and exhaust system to form $NO_2$. The remaining NO is exhausted into the atmosphere, where, in the presence of additional oxygen, it also reacts rapidly to form the more stable $NO_2$. The $NO_2$ molecules, however, can be dissociated by ultraviolet light from the sun. Additional reactions involving the ultraviolet light from the sun, nitrogen and hydrocarbon effluents, and atmospheric oxygen produce polynuclear aromatics and other components of photochemical smog.

Most of the nitrogen enters the engine as molecular $N_2$ in the air, since natural gas and the premium distillate fuels generally used in reciprocating engines contain little fuel-bound nitrogen. Oxides of nitrogen that come from $N_2$ in the air are called thermal $NO_x$. However, the heavier oils, such as residuals and crudes, contain significant quantities of nitrogen (e.g., up to 1 percent). When these fuels are used, as is occasionally done in large diesel engines, noticeable amounts of "fuel", or "organic", $NO_x$ are generated. The degree of oxidation of the nitrogen in the fuel appears to be primarily a function of its nitrogen content. For gas turbines, several researchers have found that the fraction of fuel-bound nitrogen converted to NO decreases with increasing nitrogen content, although the absolute magnitude of the NO formed increases (1,2). For example, a low nitrogen fuel (0.01 percent) may have 100 percent of the fuelbound nitrogen converted to $NO_x$, whereas a high nitrogen fuel (1.0 percent) may have only 40 percent of it reacted to $NO_x$. Consequently, when residual oil is burned in turbines, more NO is produced from fuel-bound nitrogen than from $N_2$ present in the air(3).

Similar data are not available for reciprocating engines. However, the relative magnitudes of thermal and fuel $NO_x$ are quite different for reciprocating engines than for turbines because the engines generate about 10 times more thermal $NO_x$ than do the turbines. Thus, if the fuel for an engine contained 1-percent nitrogen, and all of this were converted to oxides of nitrogen, the exhaust would contain about 5 g/hp-hr $NO_2$. By comparison, uncontrolled large-bore diesel engines emit 7.5 to 18.7 g/hp-hr when using distillate (No. 2) containing about 0.03-percent nitrogen(4).

All data presented in this document on emissions from diesel and dual fuel engines are based on operation with No. 2 distillate (diesel oil). The possible effects of any of the controls to be discussed in this chapter on the fuel-bound nitrogen are not known. Naturally, denitrification of the fuel prior to its use will reduce the potential $NO_x$ emissions, and this "control" is discussed briefly in conjunction with fuel desulfurization (see Section 4.4.13).

The rate of formation of thermal NO is a function of the residence time of the atomic nitrogen with atomic oxygen at elevated temperatures. The minimization of any or all of these three underlined variables forms the basis for almost all successfully demonstrated $NO_x$ control methods. Although exhaust treatment of the NO via catalytic converters has been proposed and tested for gasoline-fueled automobiles, research into its applicability to diesel and large natural gas engines is just reaching the experimental stage(5,6). This subject is treated in more depth later (Section 4.4.9).

### 4.1.2 Hydrocarbons (HC)

The pollutants commonly classified as hydrocarbons are composed of a wide variety of organic compounds. They are discharged into the atmosphere when some of the fuel remains unburned or is only partially burned during the combustion process.

Most unburned hydrocarbon emissions result from fuel droplets that were transported or injected into the "quench layer" during combustion. This is the region immediately adjacent to the combustion chamber surfaces, where heat transfer outward through the cylinder walls causes the mixture temperatures to be too low to support combustion.

Partially burned hydrocarbons, on the other hand, can occur for a number of reasons(7,8):

- Poor air-fuel homogeneity due to incomplete mixing prior to, or during, combustion (i.e., local zones that are too rich or too lean). The most common cause of this is improper maintenance or design of the fuel handling system.
- Incorrect air-fuel ratios in the cylinder during combustion (i.e., the entire cylinder is too rich or too lean) due to maladjustment of the engine fuel system.
- Excessively large fuel droplets (diesel engines). This is generally the result of worn, clogged, or poorly designed injectors and/or low injection pressures.
- Low cylinder temperature due to excessive cooling through the walls or early cooling of the gases by expansion of the combustion volume caused by piston motion before combustion is completed. This condition commonly occurs in engines that have faulty temperature control systems or have excessively delayed ignition.

All of these conditions can be caused by either poor maintenance or faulty design. Therefore, the lowest emissions will be achieved only by proper maintenance of engines designed specifically for low emissions.

### 4.1.3 Odor

Odor in the exhaust is primarily a problem with diesel engines. Although the smell is believed to be from some of the hydrocarbons in the exhaust, a precise relation has not yet been established between exhaust and odor(9,10,11). Considerable research has led to the belief that aldehydes may be one cause of odor in diesel (and gasoline) exhaust. Aromatic hydrocarbons and other unburned elements of fuel, as well as sulfur species, may also be a source of odor(12).

Subjective evaluation of automobile diesel odors has shown that installing sac injectors (see Section 4.5) and catalytic mufflers for hydrocarbon control also reduces odor levels(13).

Standards for odor are not being developed at this time because odor levels are low in the vicinity of large-bore engines (in part due to their relatively tall stacks), no control technology has been demonstrated specifically for odors, and no readily implemented method exists for measuring odor levels.

### 4.1.4 Carbon Monoxide (CO)

Carbon monoxide is an intermediate combustion product that appears in the exhaust when the reaction of CO to $CO_2$ cannot proceed to completion(14). This situation occurs if there is a lack of available oxygen, if the gas temperature is too low, or if the residence time in the cylinder is too short(15). The oxidation rate of CO is limited by reaction kinetics and, as a consequence, can be accelerated only to a certain extent by improvements in air-fuel mixing during the combustion process(16).

### 4.1.5 Smoke and Particulate Matter

White, blue, and black smoke may be emitted from IC engines. Liquid particulates appear as white smoke in the exhaust during an engine cold start, idling, or low load operation. These are formed in the quench layer adjacent to the cylinder walls, where the temperatures are not high enough to ignite the fuel. They consist primarily of raw fuel with some partially burned hydrocarbons and lubricating oil(17). White smoke emissions are generally associated with older gasoline engines and are rarely seen in the exhaust from diesel or gas-fueled units. They cease when the engine reaches its normal operating temperature and can be minimized during low demand situations by proper idle adjustment.

Blue smoke is emitted when lubricating oil leaks, often past worn piston rings, into the combustion chamber and is partially burned(18). Proper maintenance is the most effective method of preventing these emissions from all types of IC engines.

The exact formation mechanisms of black smoke are still not fully understood, although a number of models and theories have been offered to explain its formation in diesel engines(19,20). It is generally agreed, however, that the primary constituent of black smoke is agglomerated carbon particles (soot). These form in a two-step process in regions of the combusting mixture that are oxygen deficient(21). First the hydrocarbons decompose into acetylene and hydrogen in the high temperature regions of the cylinder. Then, when the local gas temperature drops as the piston moves down and the gases expand, the acetylene condenses and releases its hydrogen atoms. As a result, pure carbon particles are created. This mechanism of formation is associated with the low air-to-fuel ratio conditions that commonly exist at the core of the injected fuel spray, in the center of large individual fuel droplets, and in fuel layers along the walls(22). The formation of particules from this source can be reduced by designing the fuel injector to provide for an even distribution of fine fuel droplets such that they do not impinge on the cylinder walls.

Once formed, the carbon will combine with oxygen to form CO and $CO_2$ if it is still at an elevated temperature. Since the temperature of the exhaust system is too low for this oxidation to occur, soot that leaves the combustion chamber before it has had the opportunity to oxidize completely will be discharged as visible particles.

Becauase soot formation is very sensitive to the need for oxygen, its discharge is greatest when the engine is operating at rich fuel/air ratios, such as at rated power and speed. Therefore, naturally aspirated engines are likely to have higher smoke levels than turbocharged engines, which operate at leaner fuel/air ratios.

### 4.1.6 Sulfur Dioxide ($SO_2$)

Sulfur dioxide emissions are a function of only the sulfur content in the fuel rather than of any combustion variables. In fact, during the combustion process essentially all the sulfur in the fuel is oxidized to $SO_2$.[1] Manufacturers of diesel engines currently recommended that users burn only fuels which contain less than 0.5 percent sulfur(23).[2] This practice is suggested to minimize corrosion, but it also results in $SO_2$ emission levels of less than 2 g/hp-hr from most internal combustion engines.[3] Nevertherless, fuel sulfur content may range as high as 1.0 percent according to ASTM fuel specifications for No. 2 or No. 2D distillate oils(26). These distillates are the fuels most commonly burned in stationary, large-bore diesel engines. The refining of crude oil into the No. 1 and No. 2 distillate, however, removes most of the sulfur (which becomes concentrated in the residual oil). Thus, engine users generally have no difficulty obtaining low sulfur (less than 0.5 percent) distillates.

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[1] Presumably the ratio of $SO_3$ to $SO_2$ in the exhaust of IC engines is a function of the combustion process, as it is in boilers. Data are not available to clarify this question, and in any case, the total sulfur emitted is equal to the sulfur contained in the fuel.

[2] This level, which corresponds to approximately 0.7 lbm/MBtu based on heat input, is nearly one-half the value specified in the standards of performance for new oil-fired steam generators (1.2 lbm/MBtu, assuming 40 percent efficiency)(24). Existing steam plants average 3.0 to 5.0 lbm/MBtu heat input (equivalent to 8.7 to 14.4 g/hp-hr output)(25).

[3] For example, an engine firing 0.5 percent sulfur No. 2 distillate at a rate of 0.4 lb/hp-hr would emit 1.8 g/hp-hr of $SO_2$, assuming all the sulfur in the fuel is converted to $SO_2$.

Similarily, natural gas that is transported by pipeline is virtually free of sulfur. That is, the gas from some sources contains very little sulfur, and supplies which do contain sulfur (usually in the form of hydrogen sulfide) must, by law, be desulfurized before the gas can be transported in interstate commerce. This chemical desulfurization technology is routinely practiced. An "emerging" industry which makes "synthetic natural gas" from oil or coal also desulfurizes it prior to sale.

However, sulfur is generally not removed from sewage digestor gas that is burned onsite by large-bore IC engines. Nevertheless, as is discussed in Section 4.4.13, engines operating on unprocessed sewage gas emit no more than 1 g/hp-hr of $SO_2$, which is less than the $SO_2$ emission rate from engines burning 0.5 percent sulfur distillates. As a result, engines fueled with distillates and gases are already relatively low in $SO_x$ emissions. Therefore, the only potential $SO_2$ problem from reciprocating engines should be those units that are fueled with residual oil. As discussed in Section 4.4.13, the most feasible control technique for $SO_2$ reduction in an engine is the use of low sulfur fuels. Consequently, desulfurization costs are discussed in Section 8.

## 4.2 FACTORS THAT AFFECT REPORTED $NO_x$ EMISSIONS LEVELS

Ambient conditions and measurement practices have been shown to affect measured $NO_x$ emissions significantly (see Appendix C.2 and C.3). Recognizing these effects, EPA's Office of Mobile Source Air Pollution Control has required that specific ambient correction factors (to adjust measured $NO_x$

data to standard conditions of humidity and temperature) and measurement procedures be adopted by manufacturers of diesel and gasoline engines whose emissions are regulated under Title II of the Clean Air Act. For the same reasons, ambient correction factors should be used when analyzing emissions data from stationary IC engines. Likewise, consideration should be given to potential impacts on the certainty of the data due to the measurement procedures used.

To this end, Section 4.2.1 briefly reviews the effect of ambient conditions on the reported $NO_x$ data and then recommends ambient correction factors appropriate for stationary IC engines. Section 4.2.2, in turn, discusses and compares the four different measurement practices used by the nine manufacturers of large-bore stationary IC engines. Estimates of uncertainty for three of the practices (relative to the EPA procedure) are given for the manufacturers who did not follow the EPA practice. These estimates will be used to determine the uncertainty in average emission levels that were computed from data provided by the manufacturers.

### 4.2.1 Effect of Ambient Humidity, Temperature, and Pressure

EPA has adopted ambient correction factors for regulated mobile sources, based on experimental studies(27,28). These factors are used to correct observed $NO_x$ levels to values at a standard temperature (85°F) and humidity (75 grains $H_2O$/lb dry air) for heavy duty (HD) diesel engines and to correct to standard humidity, only, for both heavy and light duty gasoline vehicles(29,30). HC and CO emissions from both CI and SI engines are also sensitive to ambient humidity, temperature, and pressure. One study of diesel engines established that variations in humidity did not affect HC, CO, or smoke levels(31). Temperature variations, however, did substantially affect

HC, CO, and smoke levels, but no corrections could be generalized. In another study of automotive spark ignition engines, ambient humidity variations were shown to affect HC levels, and temperature and pressure changes to affect HC and CO emissions(32). However, no factors could be generalized for these effects.

Although the nine large-bore engine manufacturers provided ambient conditions (barometric pressure and inlet air temperature and humidity) for much of their emission data, none of them reported emission data for a given engine operated under a systematic variation of ambient conditions. Thus, at this time it is not possible to directly derive ambient correction factors for large-bore engines. Nevertheless, the response of these large-bore engines to changes in ambient conditions is anticipated to be similar to that observed in the smaller bore diesel and gasoline engines for which ambient correction factors have been developed. In fact, one large-bore manufacturer has adopted a humidity correction factor based on smaller bore engines to correct $NO_x$ emissions data from diesel and natural gas-fired engines (33). Although this manufacturer has not conducted a systematic study of the effect of ambient humidity on $NO_x$ emissions, he believes the correction factor derived from data on smaller bore engines is reasonable. This belief is based on the reasonable correlation obtained by using these correction factors for a limited number of tests with his own engines.

This section will, therefore, illustrate the effect of ambient variations on reported $NO_x$ levels, briefly review existing ambient correction factors, and recommend those suitable for application to large bore engine data. (A more detailed discussion can be found in Appendix C.2.) These ambient correction factors will then be used wherever possible in the remaining sections of this chapter to correct the data reported by the engine manufacturers.

### 4.2.1.1 Examples of Ambient Effects on $NO_x$ Emissions from IC Engines

As mentioned above, several studies have shown the effect of ambient humidity on $NO_x$ emissions(34,35,36). Figure 4-1(37) is taken from one of these studies and illustrates the variation in specific humidity each month during the year 1969 for Dearborn, Michigan. For this area, ambient humidity varied from 20 to 120 grains $H_2O$/lb dry air over the year. The effect of this variation on $NO_x$ emissions is shown in Figure 4-2(38,39) using the correction factors that have been derived from experimental work for gasoline and diesel engines. The $NO_x$ levels are shown to deviate as much as 25 percent from levels measured at standard conditions. Moreover, the corrections factors used vary significantly as well, depending on the particular study and type of emission source.

Variations in ambient temperature have also been shown to affect $NO_x$ emissions. This effect is illustrated in Figure 4-3(40,41) and ranges from 5 to 25 percent depending on the particular study. Although the effect for diesel engines is not as large as that produced by ambient humidity variations, the change in brake specific emissions (g/hp-hr) can be significant. This is particularly true for some large-bore spark ignition engines, which emit as much as 20 g/hp-hr. For these sources a 5-percent ambient correction means a change in the reported level of about 1 g/hp-hr.

Variations in barometric pressure also can be expected to affect $NO_x$ emission(42,43). Only one study, however, has evaluated this ambient variation(44). This investigation used carbureted gasoline engines and found $NO_x$ variations of as much as 40 percent due to changes in ambient pressure.

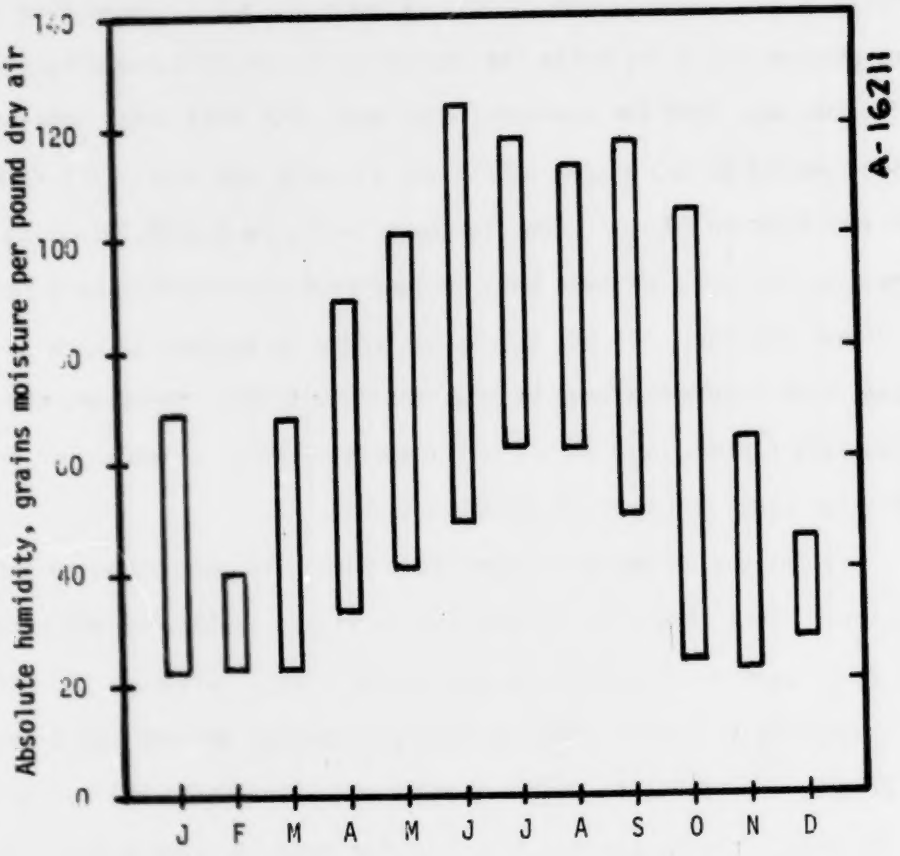
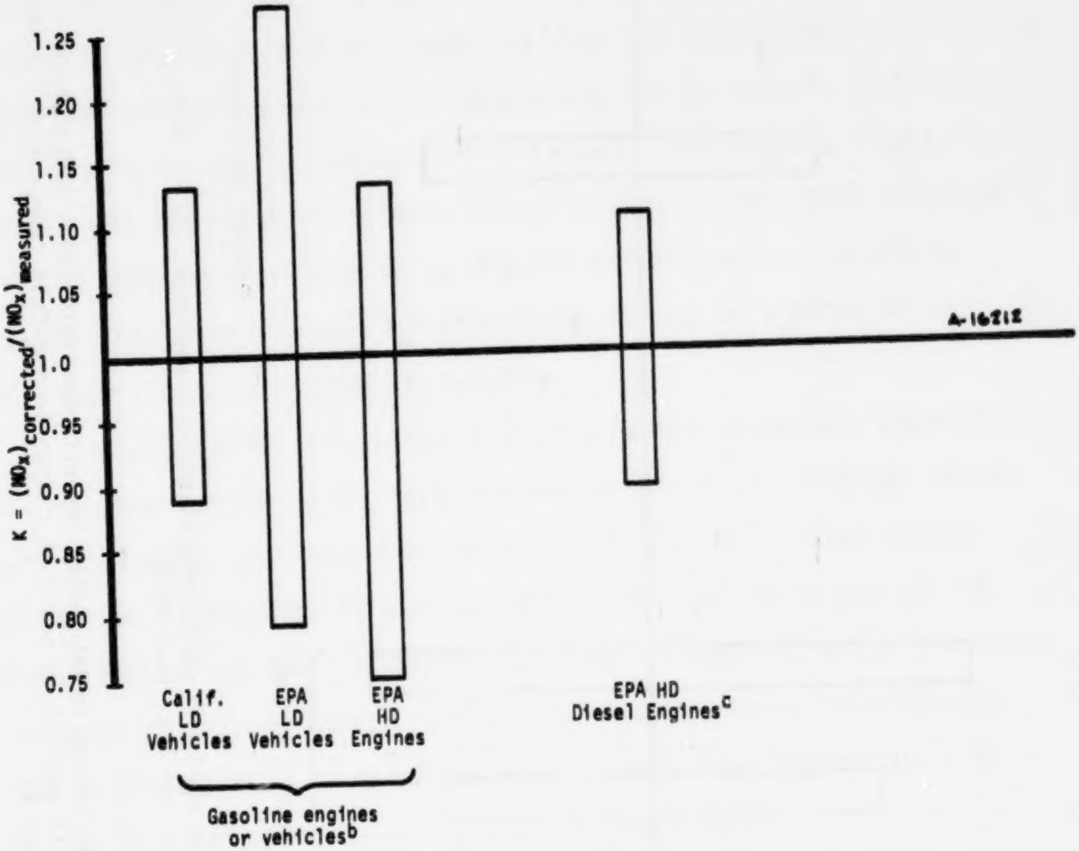


Figure 4-1. 1969 humidity range in engine test cells, Dearborn, Michigan (from Reference 37).

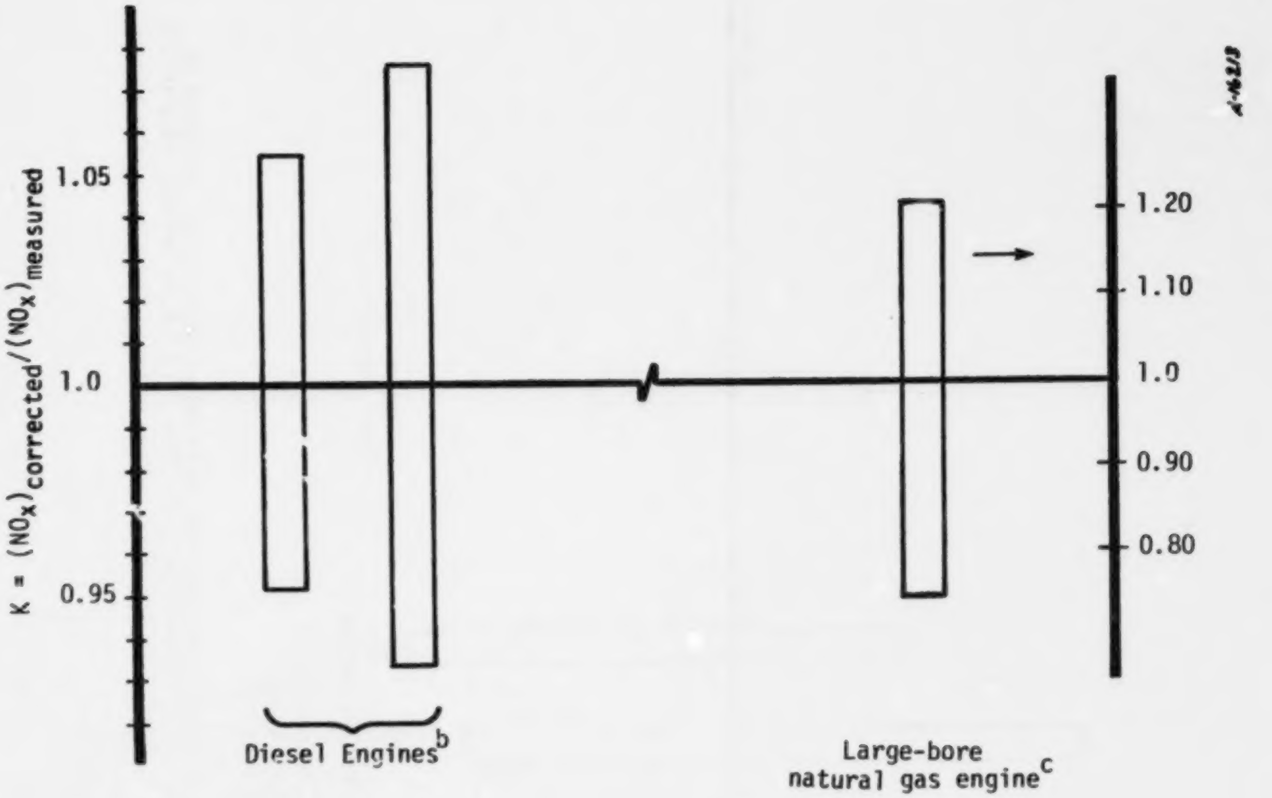


[a] Range based on humidity variation from 20 to 120 grains/lb dry air referenced to 75 grains/lb dry air standard condition. Note: $K > 1.0$ for humidity >75 grains/lb dry air.

[b] From Reference 38.

[c] From Reference 39.

Figure 4-2. Effect of ambient humidity on $NO_x$ emissions for IC engines.[a]



[a] Based on range of ±30°F from standard temperature 85°F. Note: $K > 1$ for temperatures less than 85°F.

[b] Reference 40.

[c] Based on change in $NO_x$ emissions with manifold inlet temperature. Assume change in ambient is equal to change in manifold temperature. From Reference 41.

Figure 4-3. Effect of ambient temperature on $NO_x$ emission levels from IC engines.[a]

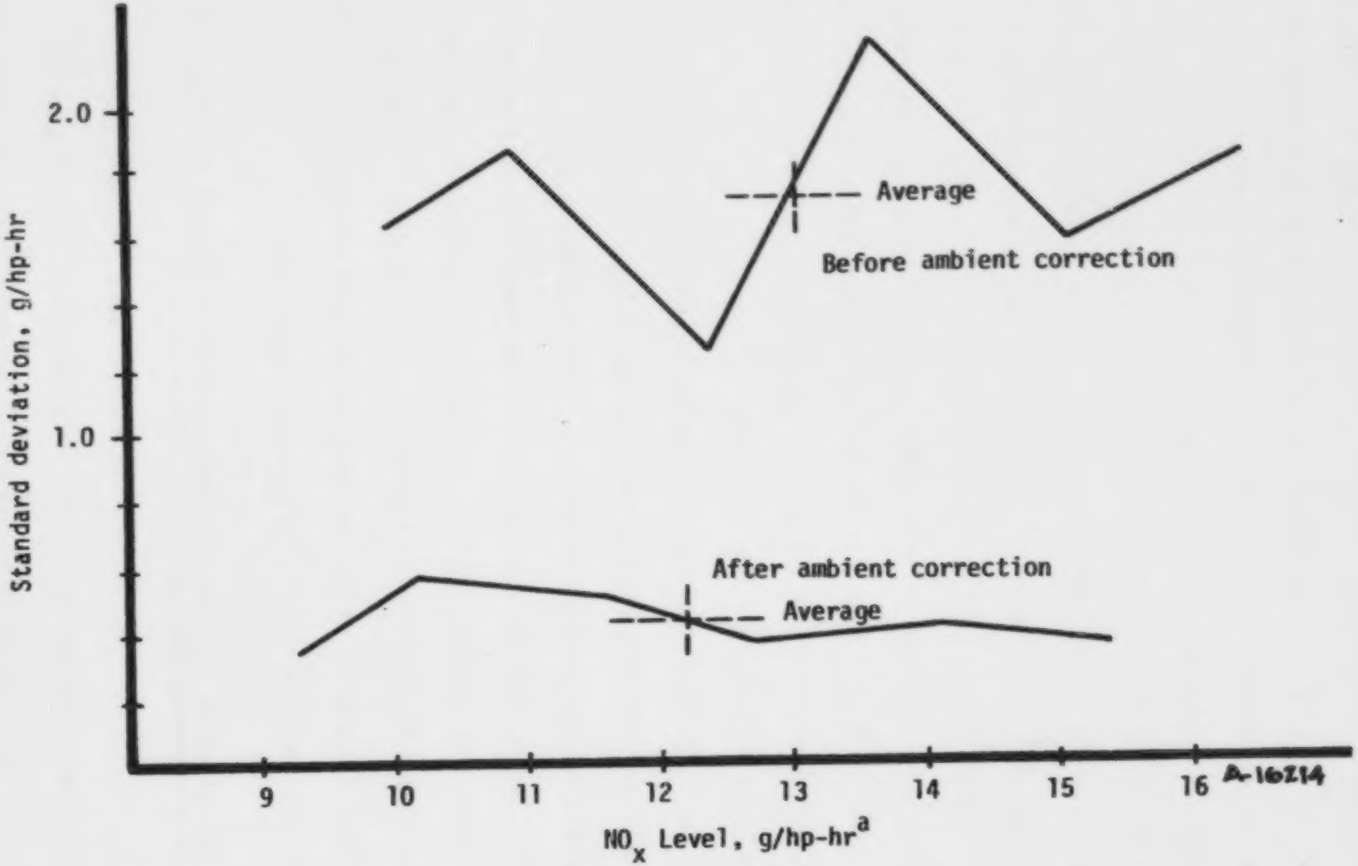
These changes were attributed largely to variations in A/F ratio in the carbureted gasoline engines. However, a correction factor for this effect could not be derived for the different carbureted engines because each engine's emissions response to changes in barometric pressure was too inconsistant to generalize a correction.

Despite the lack of a quantifiable correction for changes in ambient pressure, several studies have shown that scatter in emissions data taken at different ambient humidities and temperatures can be reduced, significantly, by applying the appropriate correction factors. For example, Figure 4-4(45) illustrates the reduction in data scatter that is achieved with emissions from HD gasoline engines by correcting for humidity only. The average standard deviation from the six engines was reduced by a factor of about six after applying a correction for humidity.

Figure 4-5(46) illustrates a similar result for ambient temperature and humidity correction of diesel engines emissions. The average scatter after correction was reduced to about one third of the scatter before correction. Note, that in absolute terms, the reduction in scatter for diesel engines was smaller (approx. 0.5 g/hp-hr) compared to the reduction for gasoline engines (approx. 1.5 g/hp-hr). Both studies, nevertheless, indicate that scatter in emissions data can be reduced significantly by correcting observed $NO_x$ levels to standard conditions.

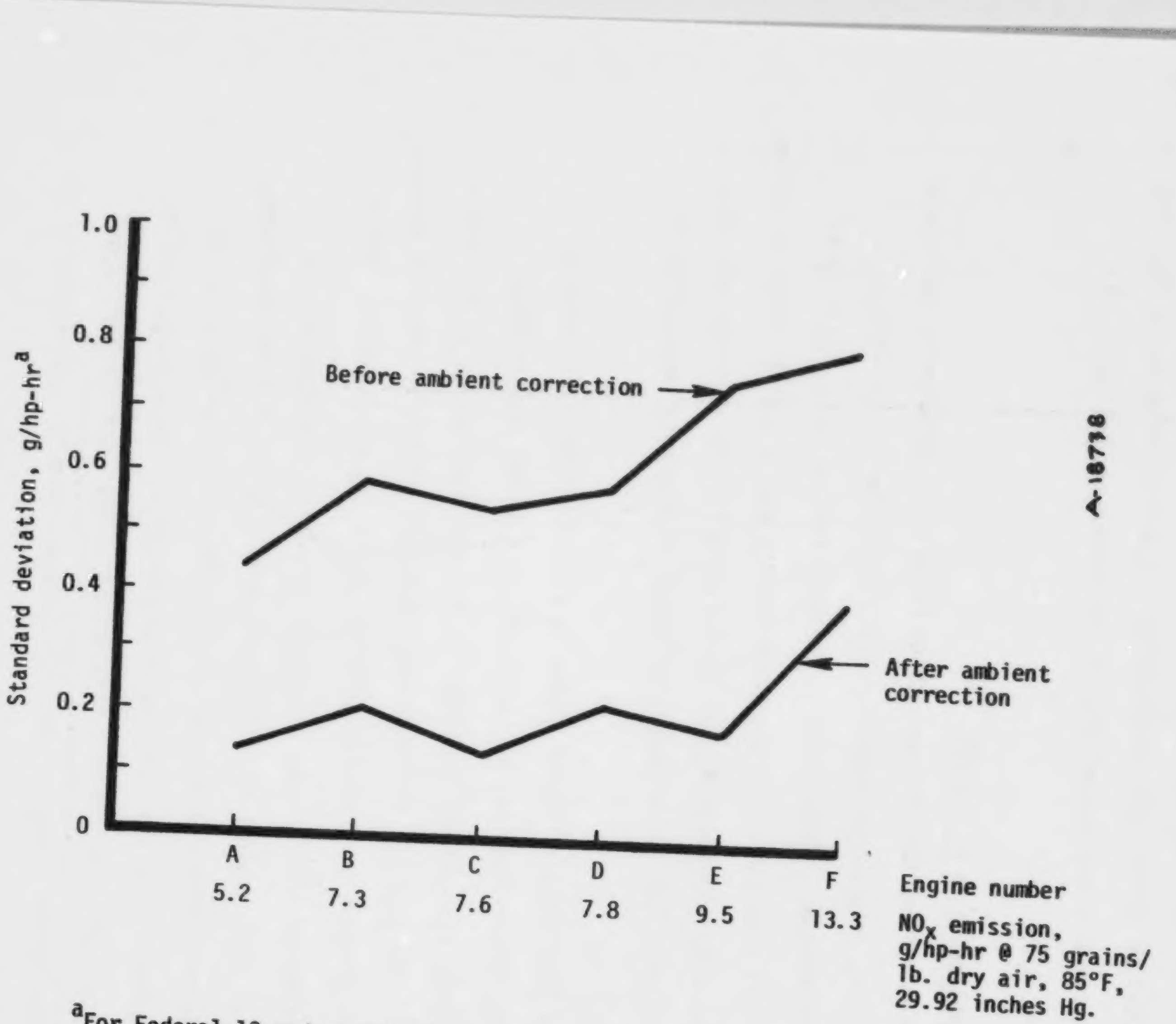
#### 4.2.1.2 Selection of Existing Ambient Correction Factors for Application to Large-Bore IC Engines

All existing ambient correction factors were reviewed that potentially could be applied to adjust data from large-bore engines. A detailed description of this review is presented in Appendix C.2. Candidate factors are presented below. This discussion is divided into two sections depending



[a] Average level (Federal Test Cycle Composite) for four ambient humidities; 30, 45, 75, 120 grains/lb dry air.

Figure 4-4. Effect of humidity on emissions scatter for six HD gasoline engines (Reference 45).



[a] For Federal 13 mode composite cycle for HD diesel engines.

[b] Ambient humidity varied from 35 to 125 grains/lb dry; temperature from 70 to 115°F.

Figure 4-5. Effect of humidity and temperature on emissions scatter for six HD diesel engines[b] (Reference 46).

on whether the correction factor can be applied to data from (1) spark ignition (SI) or (2) compression ignition (CI) engines. Factors that have been selected to correct both uncontrolled and controlled emission data from the nine large-bore engine manfacturers are summarized following this discussion.

## Factors Applicable to SI Engines

Three ambient humidity correction factors are potentially applicable to large bore, natural gas fueled engines, particularly four-stroke, carbureted versions. These factors are summarized in Table 4-1(47,48,49) Figure 4-6 is a comparison of the three factors over a typical range of ambient humidities. Note that only one of the three factors is at a constant load factor (Equation (2b)); the others are based on composite test cycles for vehicles.

As Figure 4-6 indicates, there is a considerable difference in correction factor depending on the study. All of the studies show, nevertheless, that ambient humidity has a significant effect on $NO_x$ level. The result for Equation (3), based on light-duty automotive gasoline vehicle, shows the greatest sensitivity to variations in ambient humidity. These varied responses of $NO_x$ level to changes in ambient humidity are not unexpected since engines react differently to changes in inlet conditions. Their response generally depends on their A/F ratio, fuel metering and distribution system, and ignition characteristics. Since large-bore IC engines typically operate at a constant rated load, the constant load correction factor (Equation (2b)) has been selected for application to the reported data.

TABLE 4-1. AMBIENT HUMIDITY CORRECTION FACTORS FOR SI ENGINES

| Equation No. | Correction Factor[a] | Reference |
|---|---|---|
| (1) | $K = 0.634 + 0.00654(H) - 0.0000222(H)^2$ Composite Factor (9 Mode Federal HD Gasoline Test Cycle) | 47 |
| (2a) | $K = 0.796 + 0.175(H/100) + 0.129(H/100^2)$ Composite Factor (Federal Test Cycle, LD Gasoline Vehicles) | 48 |
| (2b) | $K = 0.844 + 0.151(H/100) + 0.075(H/100)^2$ 50 mph, Constant Load | 48 |
| (3) | $K = 1/(1 - 0.0047(H-75))$ Composite Factor (Federal Test Cycle, LD Gasoline Vehicles) | 49 |

[a] H = specific humidity in grains $H_2O$/lb dry air.

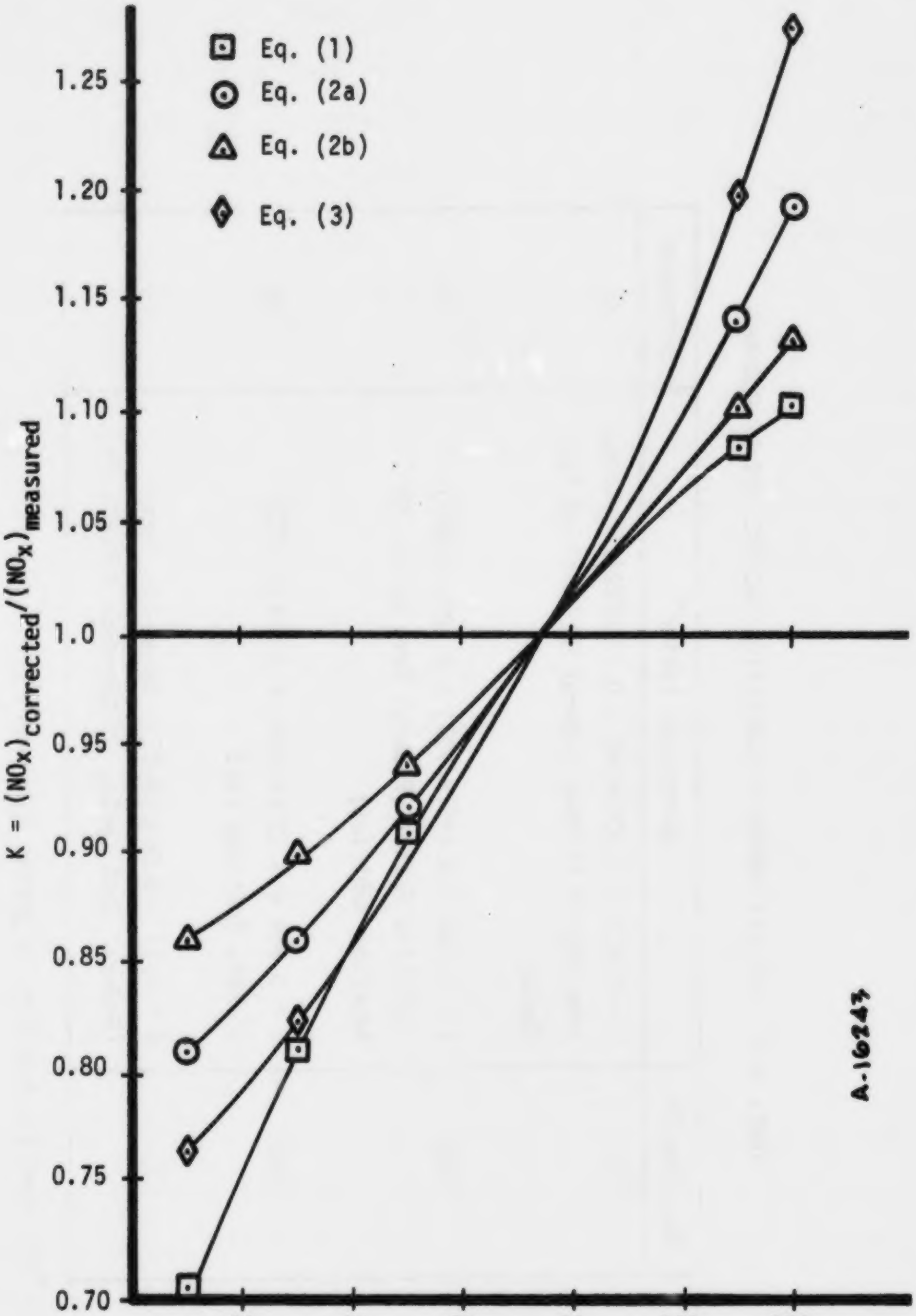


Figure 4-6. Comparison of SI ambient humidity correction factors.

Previous investigators have been unable to establish an ambient temperature correction factor for spark ignition IC engines because various automotive engines respond quite differently to inlet air temperature variations (see Appendix C.2). A limited amount of data exists for large IC engines that show the variation in $NO_x$ emissions with ambient temperature, or manifold air temperature for turbocharged units. Figure 4-7(50) illustrates the change in $NO_x$ level with a change in manifold air temperature for a large-bore, four stroke per cycle, turbocharged (4-TC) gas engine. This response indicates approximately a 1-percent change in $NO_x$ level per °F change in manifold air temperature. (A change in manifold air temperature is nearly equivalent to the same change in ambient temperature.)

Figure 4-8(51) indicates the response of $NO_x$ emissions with changes in ambient air temperature for large-bore, blower-scavenged gas engines. These results show approximately a 2-percent change in $NO_x$ per °F change. Both Figures 4-7 and 4-8 indicate that $NO_x$ emissions from large-bore engines are very sensitive to ambient temperature variations. Therefore the results of Figure 4-7 will be applied to all turbocharged gas engine data, and the results of Figure 4-8 will be applied to all nonturbocharged gas engine data.

## Factors Applicable to IC Engines

A survey of the literature established two sources that have reported ambient correction factors for truck-size diesel engines. In the study by Krause, et. al., a factor was developed that included the effects of temperature and humidity(52). The results of this study were subsequently adopted by the EPA for mobile heavy duty diesel engines. The other study was conducted by the Coordinating Research Council (CRC) and only investigated

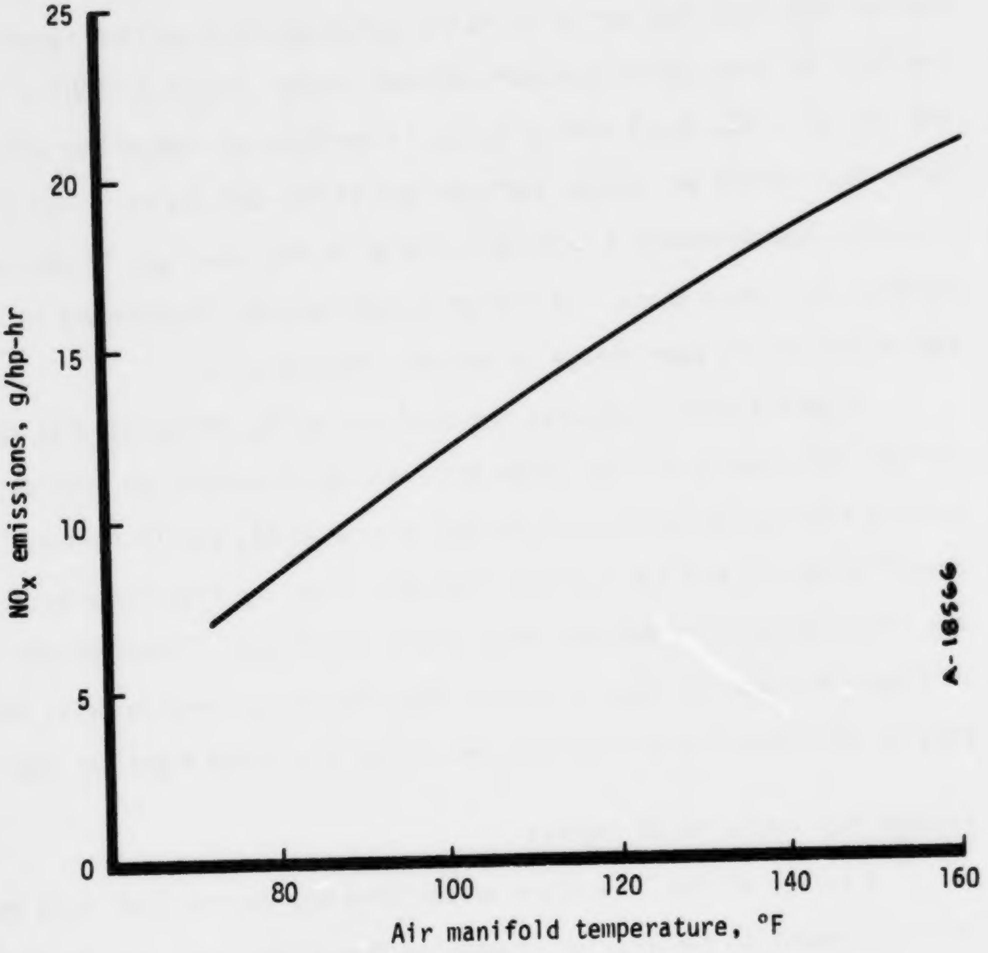


Figure 4-7. Effect of manifold air temperature on a large-bore 4-TC engine (Reference 50).

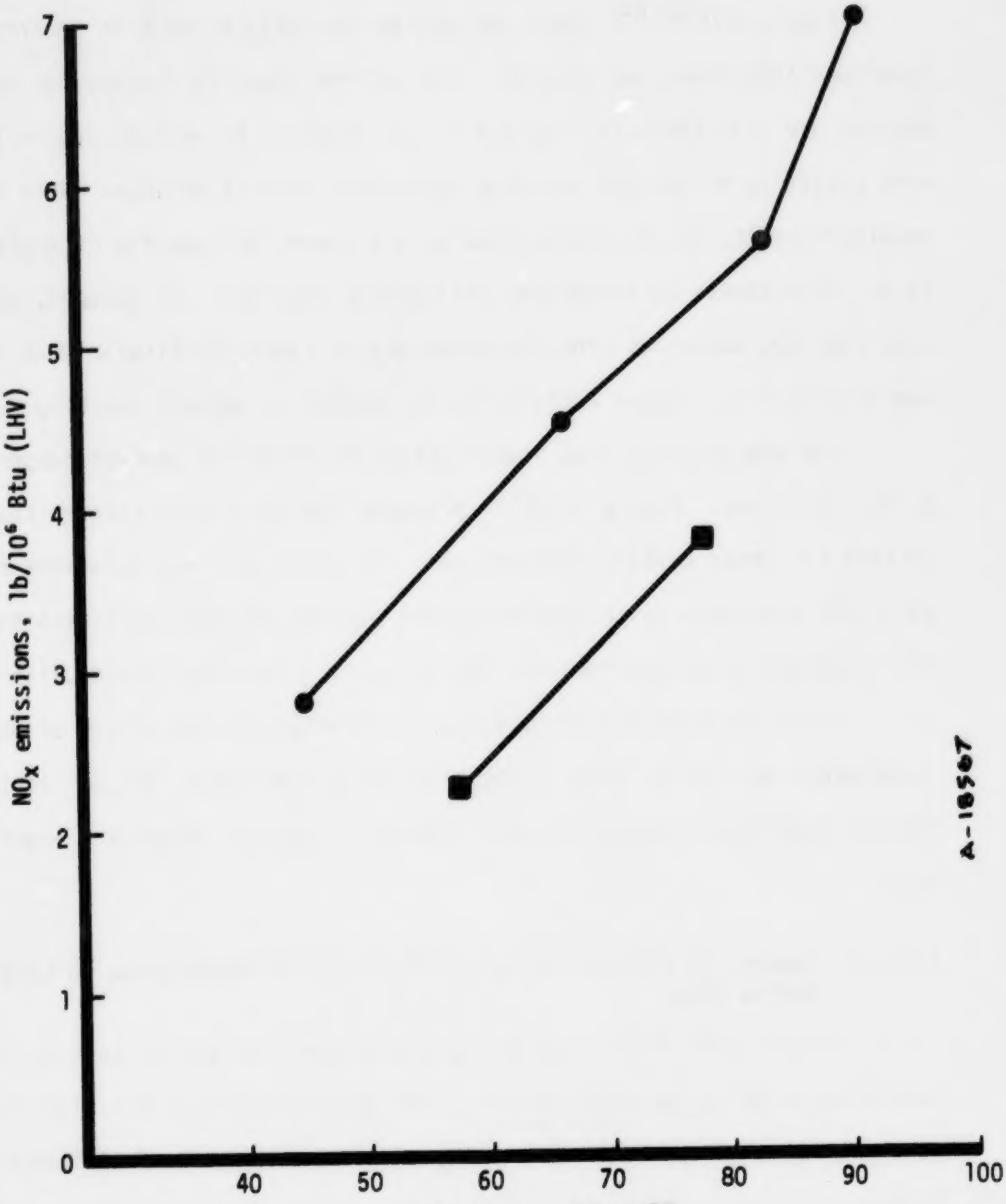


Figure 4-8. Test results of NO$_x$ emissions versus intake air temperature for two blower-scavenged gas engines (Reference 51).

the effects of ambient humidity(53). (A more detailed discussion of both studies can be found in Appendix C.2.)

Figure 4-9(54,55) shows the ambient humidity correction factors developed from these two studies. The ambient humidity factors for SI engines are also shown in Figure 4-9. In general, SI engines appear to be more sensitive to ambient humidity variations than CI engines. Note that the results from the two CI engine studies are shown for specific CI engine types (e.g., four-stroke turbocharged, aftercooled engines). In general, the data show that $NO_x$ emissions from different engine types, particularly at low humidity levels, respond differently to changes in ambient humidity.

The Krause study also investigated the effect of ambient temperature on $NO_x$ emissions. Figure 4-10(56) presents the correction factors that were derived for these smaller bore engines. It shows that $NO_x$ emissions from naturally aspirated and blower-scavenged engines are more sensitive to inlet air temperature changes than are the emissions from aftercooled units.

Since the Krause study systematically examined the effect of both temperature and humidity for a number of CI engine types, his correction factors have been selected for application to similar large-bore engine types.

#### 4.2.1.3 Summary of Ambient Correction Factors for Application to Large-Bore Engine Data

Table 4-2(57,58,59) summarizes the ambient correction factors that have been selected for application to the data reported by the nine large-bore engine manufacturers. Note that, with the exception of SI temperature factors, all the corrections are based on studies of smaller bore automotive engine types. The corrections for CI engines are given for specific engine

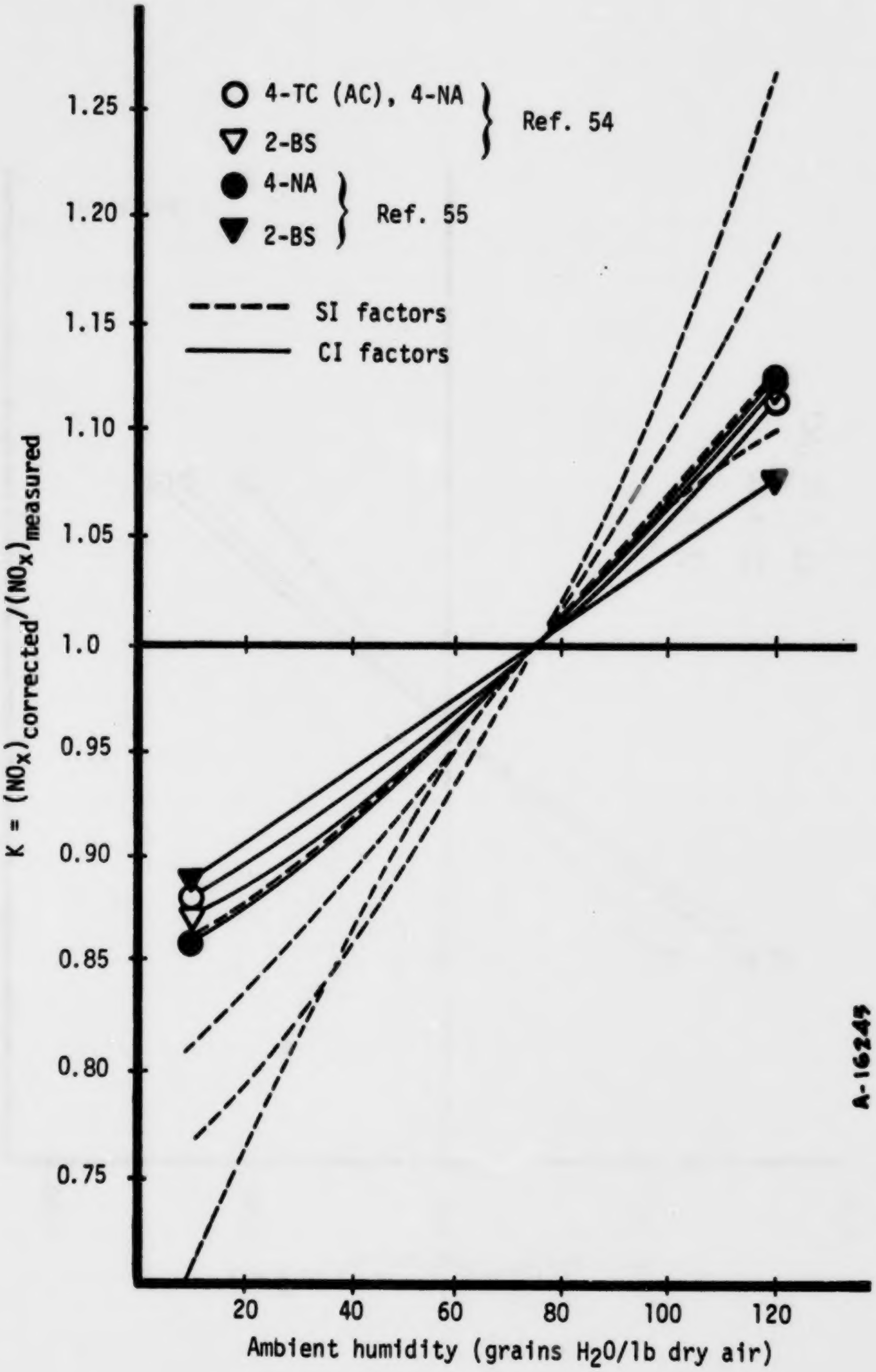


Figure 4-9. Comparison of CI and SI ambient humidity correction factors.

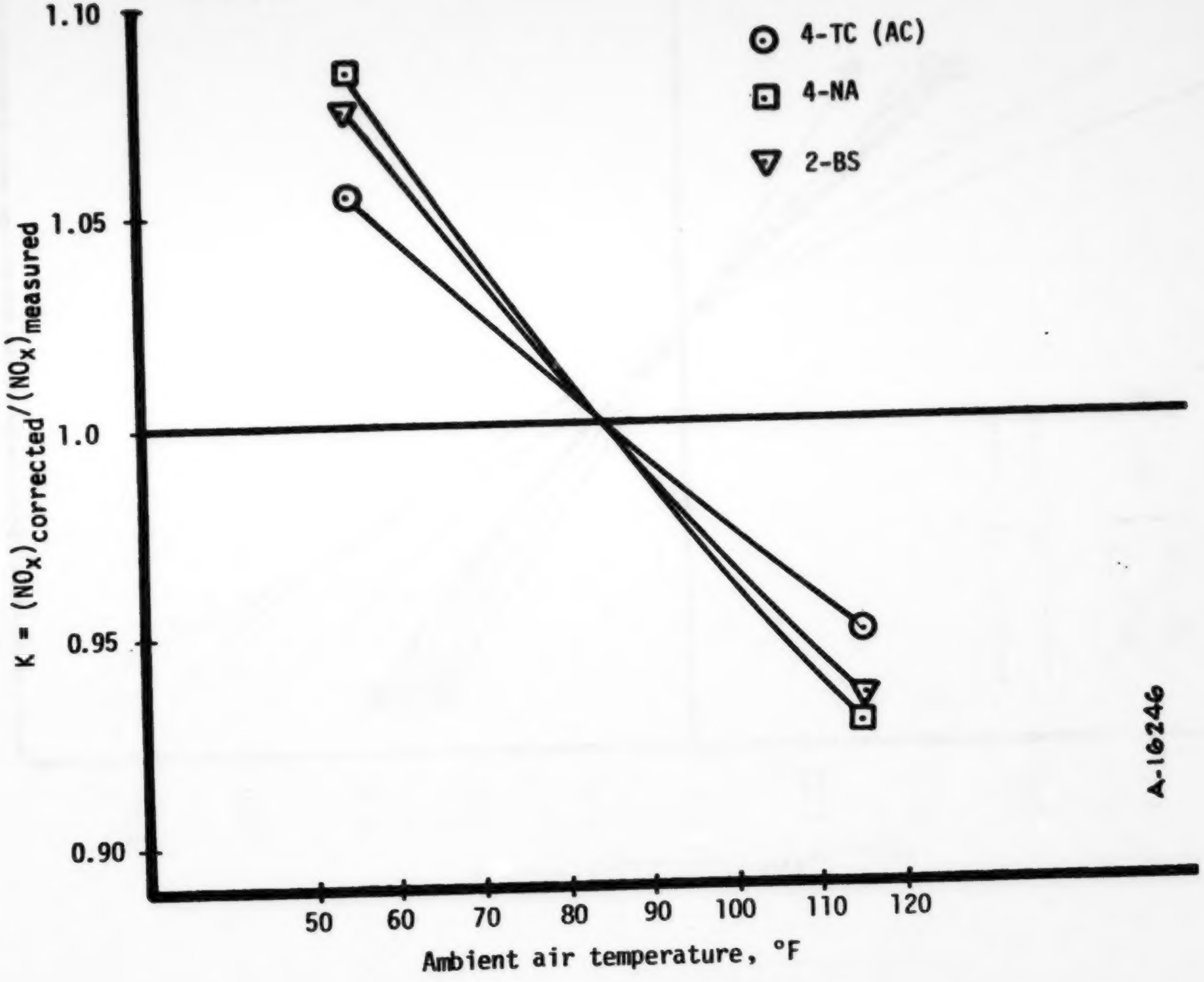


Figure 4-10. Correction factors for temperature for CI engines (Reference 56).

TABLE 4-2. EXISTING IC ENGINE AMBIENT CORRECTION FACTORS FOR APPLICATION TO LARGE-BORE ENGINES

| Fuel | Correction Factor[a] | | | Comments |
|---|---|---|---|---|
| Diesel & Dual Fuel (CI)<br>Humidity and Temperature Correction | $K = 1/(1+A(H-75) + B\ (T-85))$<br>H = observed humidity, grains $H_2O$/lb dry air<br>T = observed inlet, air temperature, °F | | | Rated load correction for humidity and temperature (from Ref. 57). |
| | Type | A | B | |
| | 2-BS | -0.00242 | 0.00235 | |
| | 4-NA | -0.00231 | 0.00255 | |
| | 4-TC, AC | -0.00231 | 0.0017 | |
| Natural Gas (SI) | | | | |
| Humidity | $K = 0.844 + 0.151 \left(\frac{H}{100}\right) + 0.075 \left(\frac{H}{100}\right)^2$ | | | Rated load correction for humidity only (from Ref. 58) |
| Temperature | BS $K = 1 - (T - 85)(0.017)$<br>TC $K = 1 - (T - 85)(0.010)$ | | | Ambient temperature correction only (from Ref. 59) |

[a] $NO_x$ corrected = (K) $NO_x$ observed

types. In addition all the factors are based on rated load conditions, since large bore engines typically operate at, or near, rated load.

Although ambient humidity and temperature variations can significantly affect the $NO_x$ emissions that are measured from a particular engine, these variations, in general, are not responsible for the large variations in uncontrolled emissions that were reported for similar engine types by different engine manufacturers. The other sources of data variability (largely measurement practices and design differences among models) are discussed in Sections 4.2.2 and 4.3.

### 4.2.2 Effect of Measurement Practices

Previous studies have shown that sampling instrumentation and procedures have a large effect on emission levels. For example, a series of studies conducted by the Coordinating Research Council (CRC) indicated that uncertainties in the measurement of $NO_x$ levels can range as high as 40 percent(60). This conclusion was based on the standard deviation of measurements reported by different laboratories for the same emission source, expressed as a percentage of the mean emission level. CRC concluded that this uncertainty could be attributed largely to poor calibration and measurement procedures. The EPA, then in cooperation with CRC, showed that these uncertainties could be reduced to less than 5 percent using a specific set of procedures(61). Since then, EPA has proposed that these procedures be used to certify mobile, heavy-duty diesel and gasoline engines starting with the 1979 model year. The following paragraphs will briefly discuss the measurement practices of each of the nine large-bore engine manufacturers who reported emissions data. (Additional details regarding these practices can be found in Appendix C.3.) Then uncertainties for each manufacturer's practice will be estimated relative to the proposed EPA procedure. These uncertainties will be used in Section 4.3 to establish upper and lower bounds for estimated average emission levels from large-bore IC engines.

Table 4-3(62,63,64,65) indicates which of the four measurement practices was used by each of the manufacturers. Note that two manufacturers, Alco and Ingersoll-Rand, used what is essentially the EPA procedure. For the purpose of this discussion the EPA procedure will serve as a reference for comparison with the other three procedures, since it is believed to be the most accurate.

Figure 4-11 illustrates each of the four procedures schematically, and Table 4-4 summarizes the sources of error for the DEMA, SAE, and EMD practices. (A more detailed discussion of these procedures can be found in Appendix C.3.) The primary shortcoming of these practices is their failure to adequately define instrument performance and sample transfer procedures. Unheated sample lines, inappropriate water removal devices, system leaks, and failures of the convertor in the chemiluminescent instrument all lead to errors in the measurement of $NO_x$ in the sample gas. The use of NDIR instruments (SAE/EMD practices), can lead to overstated values of $NO_x$ emissions due to interferences resulting from the presence of water vapor in the detector cell of the NDIR instrument. Considered together, these sources of error can cause a large uncertainty in reported $NO_x$ levels. Figure 4-12 illustrates the overall uncertainty for data reported by each of the nine engine manufacturers. (Again, a more detailed discussion of these uncertainties can be found in Appendix C.3.) Note that manufacturers using the SAE or EMD procedure could experience uncertainties of ±20 percent. Manufacturers using the DEMA practice, in contrast, are more likely to experience understated (5 to 15 percent) $NO_x$ levels due to a loss of $NO_x$

TABLE 4-3. LARGE-BORE ENGINE MANUFACTURERS MEASUREMENT PRACTICES

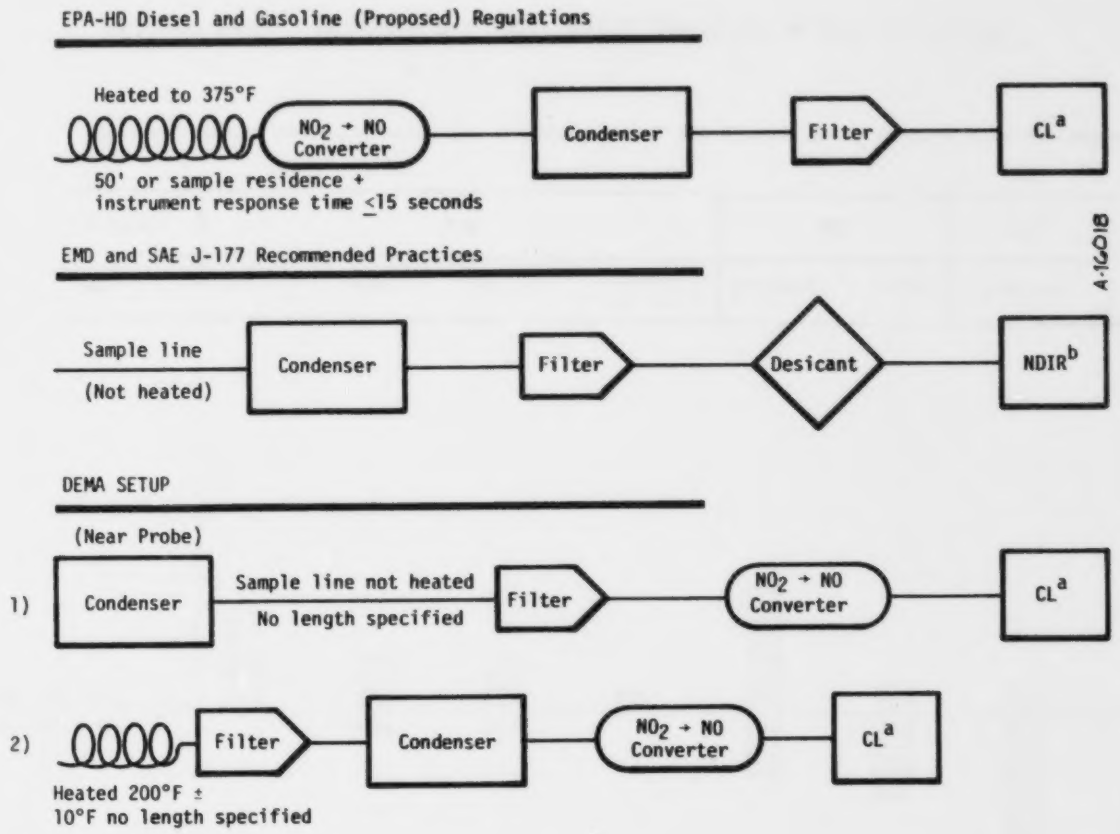
| Manufacturer | Measurement Practice | | | |
|---|---|---|---|---|
| | EPA[a] | DEMA[b] | SAE[c] | EMD[d] |
| Alco | X | | | |
| Caterpillar | | | X | |
| Colt | | X | | |
| Cooper Energy | | X | X | |
| DeLaval | | X | | |
| ElectroMotive (GMC) | | | | X |
| Ingersoll-Rand | X | | | |
| Waukesha | | | X | |
| White Superior (Div. Cooper) | | X | | |

[a] EPA's proposed practice for 1979 Heavy Duty Diesel and Gasoline Engines(62).

[b] Diesel Engine Manufacturers Association (DEMA) Exhaust Emission Measurement Procedure for Low and Medium Speed Engines(63).

[c] Society of Automotive Engineers (SAE) Recommended Practice J177a(63).

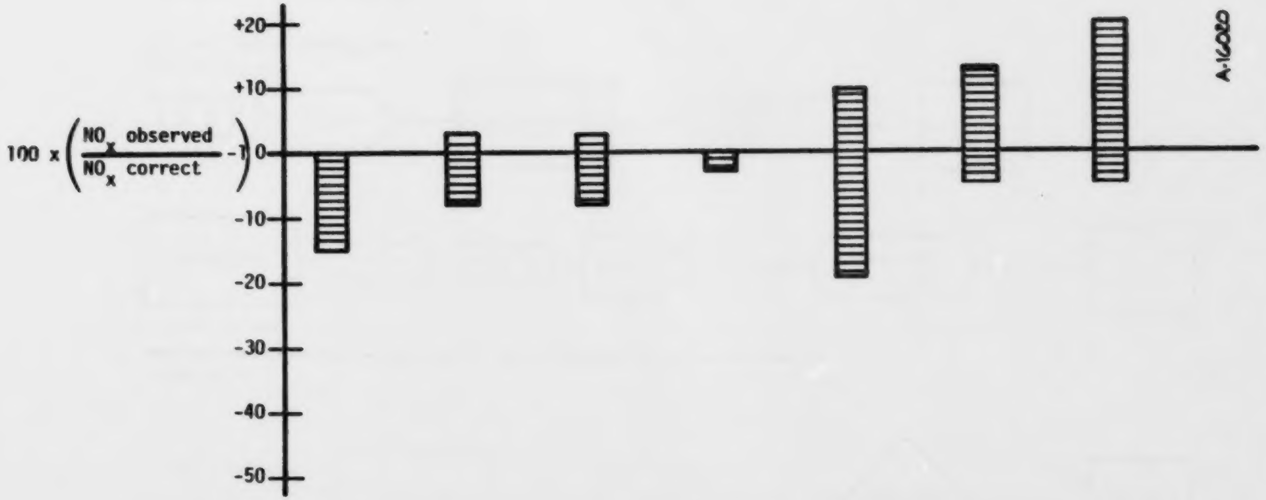
[d] ElectroMotive Division of General Motors Corporation Practice(65).


EPA-HD Diesel and Gasoline (Proposed) Regulations
Heated to 375°F
$NO_2 \rightarrow NO$ Converter
Condenser
Filter
CL[a]
50' or sample residence + instrument response time ≤15 seconds
A-16018
EMD and SAE J-177 Recommended Practices
Sample line
(Not heated)
Condenser
Filter
Desicant
NDIR[b]
DEMA SETUP
(Near Probe)
1)
Condenser
Sample line not heated
No length specified
Filter
$NO_2 \rightarrow NO$ Converter
CL[a]
2)
Filter
Condenser
$NO_2 \rightarrow NO$ Converter
CL[a]
Heated 200°F ± 10°F no length specified

[a] CL – Chemiluminescent analyzer

[b] NDIR – Nondispersive infrared analyzer

Figure 4-11. Sampling practices.



| Mfg | Colt | Cooper | Delaval | White Superior | Waukesha | Cooper | GMC/EMD |
|---|---|---|---|---|---|---|---|
| Practice | DEMA | | | | SAE | | EMD |

[a] Does not include potential error due to system leaks, and assumes chemical drier is conditioned.

Figure 4-12. Measurement uncertainty relative to EPA procedure.[a]

TABLE 4-4. SOURCES OF ERROR FOR DEMA, SAE, EMD EMISSION PRACTICES

| DEMA (CL) | EMD (NDIR/NDUV)/SAE (NDIR) |
|---|---|
| • Unheated sampling lines permitted<br>• No specification of sample residence time in sampling line, or system response time<br>• Leak checks are not specified<br>• Water removal device can be located at analyzer, but before $NO_2 \rightarrow NO$ absorption in water trap<br>• Chemical driers permitted<br>• No instrument specifications<br>• Converter efficiency checks not specified<br>• No calibration procedures specifed<br>• Calibration and span gas specifications not defined | • Unheated sample lines<br>• No sample residence or system response time specified<br>• Leak checks not specified<br>• Allow chemical drier<br>• Calibration procedures not specific (e.g., what constitutes out-of-calibration, how calibration points are curve fit, etc.)<br>• Calibration and span gas blends and dilutents not specified by SAE. EMD has "own" specifications. |

during the transfer of the sample gas to the relatively interference-free chemiluminescent analyzer. These estimates of uncertainty will be used to place upper and lower bounds on the average uncontrolled emission levels computed in Section 4.3.

## 4.3 UNCONTROLLED EMISSION LEVELS

This section presents data on uncontrolled emissions from large-bore engines. Average uncontrolled $NO_x$ emissions from these engines, weighted according to sales, were derived from data supplied by manufacturers. By applying a specified degree of $NO_x$ control to these average uncontrolled emission levels, potential controlled (regulated) emission levels can be established. The degrees of $NO_x$ control that can be applied are identified in Chapter 6, which summarizes demonstrated alternative controls.

This approach to setting the standards requires an adequate sample of emission data for each manufacturer's engine. Section 4.3.1 discusses the current data base, and shows that emissions data have been reported for about 80 percent of all the large-bore engine models manufactured to burn diesel, dual fuel, and natural gas. This large existing data base is representative of all the engines to be affected by standards of performance.

Section 4.3.2 presents the uncontrolled emissions data for diesel, dual fuel, and natural gas engines, and examines the sources of variations in these data. Differences in the ambient conditions (temperature and humidity) and procedures for measuring the emissions account for only small variations in the data. The largest source of data variations is differences in engine design. These differences and their effect on $NO_x$ emission levels are discussed in Section 4.3.3.

Because of these differences, a method is needed to characterize uncontrolled emissions from each of the three fuels for which standards of performance will be proposed. In Section 4.3.4, representative uncontrolled $NO_x$ levels for each fuel are determined by weighting each manufacturer's data (corrected for ambients where possible). Weighting is based on the percentage of total horsepower sold by each mnaufacturer during the past 5 years, and the weighted levels are bounded by estimates of measurement uncertainty, based on each manufacturer's procedures.

### 4.3.1 Existing Data Base

The extent of the current emissions data base is illustrated in Table 4-5, which shows the number of large-bore manufacturers who produce engine models within the diesel, dual fuel, and natural gas categories. The second row of the table shows the number of models produced for each engine type (e.g., 2-BS, 2-TC, 4-NA, and 4-TC). The lower two rows show the number of models within each fuel and engine type category that contain (1) uncontrolled and (2) controlled emissions data. Uncontrolled emissions data are available from every manufacturer of large-bore engines, although a few manufacturers have not conducted tests to reduce $NO_x$ emissions from their engines.

In general, as shown in the last column of Table 4-5, there are uncontrolled emissions data for about 80 percent of the models produced for each fuel category. The current data base contains more data than for those models listed in Table 4-5, since there are data for several engines of the same model for some manufacturers. This additional data is useful in determining differences in emissions data from engines of the same, and different manufacturers, as discussed in Section 4.3.3. Thus, a substantial data base exists for characterizing uncontrolled emissions from diesel, dual fuel, and natural gas engines.

## TABLE 4-5. EXTENT OF EXISTING DATA BASE

| Fuel | Diesel | | | | Dual Fuel | | Gas | | | | Overall | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Strokes | 2 | | 4 | | 2 | 4 | 2 | | 4 | | | | |
| Air Charging | BS | TC | NA | TC | TC | TC | BS | TC | NA | TC | Diesel | Dual Fuel | Gas |
| No. of manufacturers | 2 | 2 | 1 | 5 | 1 | 4 | 2 | 2 | 4 | 6 | 6 | 4 | 7 |
| No. of models[a] | 2 | 2 | 3 | 8 | 1 | 5 | 2 | 5 | 5 | 14 | 15 | 6 | 28 |
| No. of models with uncontrolled data | 2 | 2 | 1 | 6 | 1 | 4 | 2 | 6 | 4 | 10 | 11 | 5 | 23 |
| No. of models with controlled data | 2 | 2 | 1 | 5 | 1 | 3 | 2 | 6 | 3 | 7 | 10 | 4 | 18 |

[a] A model is a group of engines that share the same fuel, air charging, strokes/cycle, manufacturer, bore, and stroke.

### 4.3.2 Uncontrolled Emission Levels

Uncontrolled emissions of $NO_x$, CO, and HC (and nonmethane HC where measured) are shown on Figures 4-13 to 4-15. On each figure the data is plotted separately for each fuel (diesel, dual fuel, and gas), and is differentiated by engine type (i.e., 2-BS, 2-TC, 4-NA, and 4-TC). Since in general, the CO and nonmethane HC levels from these engines are considerably lower than the limits that apply to mobile vehicles and engines,[4] this section is concerned primarily with $NO_x$ emissions. The effects of $NO_x$ reduction techniques on CO, HC, and smoke emissions are discussed in Section 4.4.12. Figures 4-13 through 4-15 show that uncontrolled emission levels vary considerably within each category of fuel and engine type. In Figure 4-13, fuel consumption, particularly for diesel and dual-fuel engines, remains relatively constant despite wide variations in $NO_x$ levels among all engine types. Since both $NO_x$ emissions and thermal efficiency increase as cylinder temperature increases, efficient engines (low fuel consumption) would be expected to show high $NO_x$ emission rates. As shown in Figure 4-16, this is not the case among all engines of one fuel. $NO_x$ levels and four- and two-stroke diesel engines shown on Figure 4-16(a), however, appear to increase as fuel consumption decreases, but other trends are not apparent for other fuels and engine types. Although other design features (e.g., manifold air temperature, air-to-fuel ratio, speed, torque, etc.) are probably more

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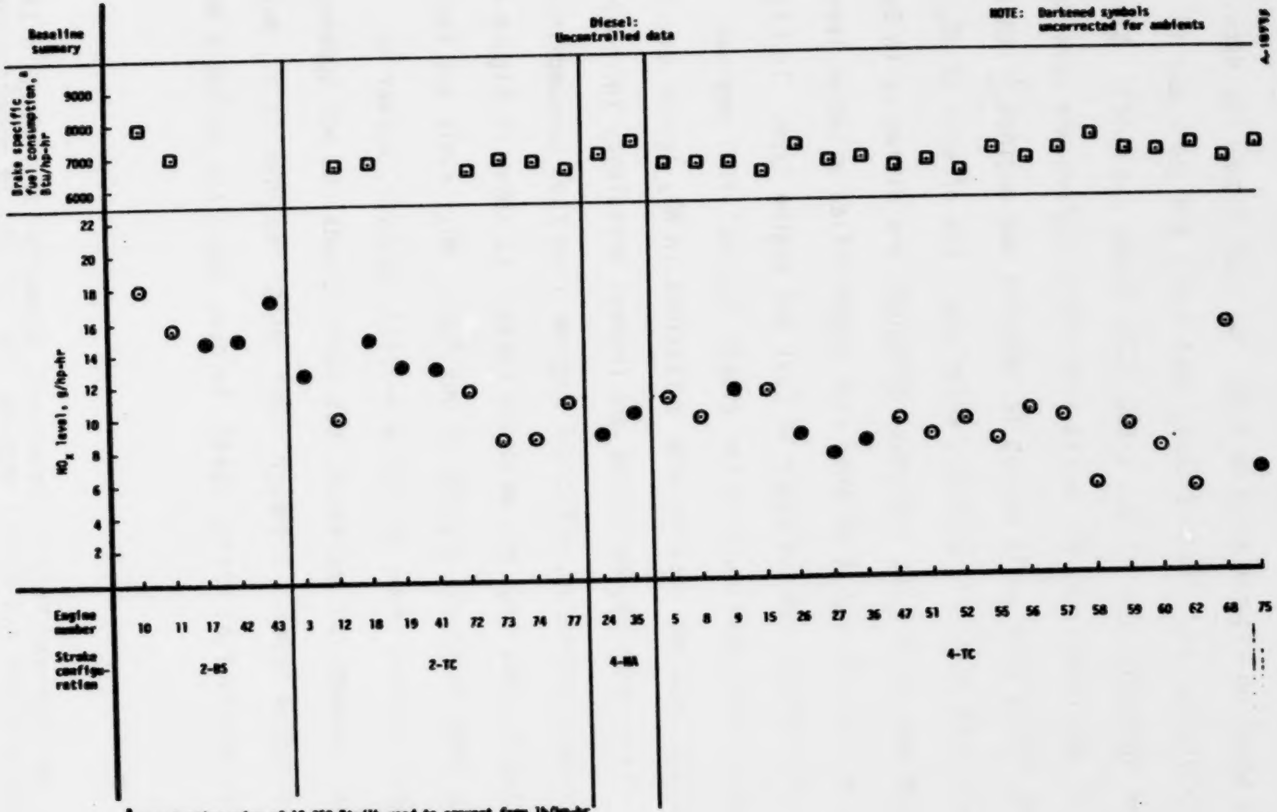
[4] For example, the proposed Federal Government standards beginning in 1979 for heavy-duty gasoline and diesel engines are 1.5 g/hp-hr hydrocarbon (HC), 25 g/hp-hr carbon monoxide (CO), and 10 g/hp-hr hydrocarbon plus oxides of nitrogen (HC + $NO_x$). California regulations for 1977-1978 heavy duty vehicles (greater than 6,000 lbs) are 1.0 g/hp-hr HC, 25 g/hp-hr CO, and 7.5 g/hp-hr $NO_x$.



ªLower heating value of 18,350 Btu/lb used to convert from lb/hp-hr

Figure 4-13(a). Uncontrolled $NO_X$ emissions from diesel engines.

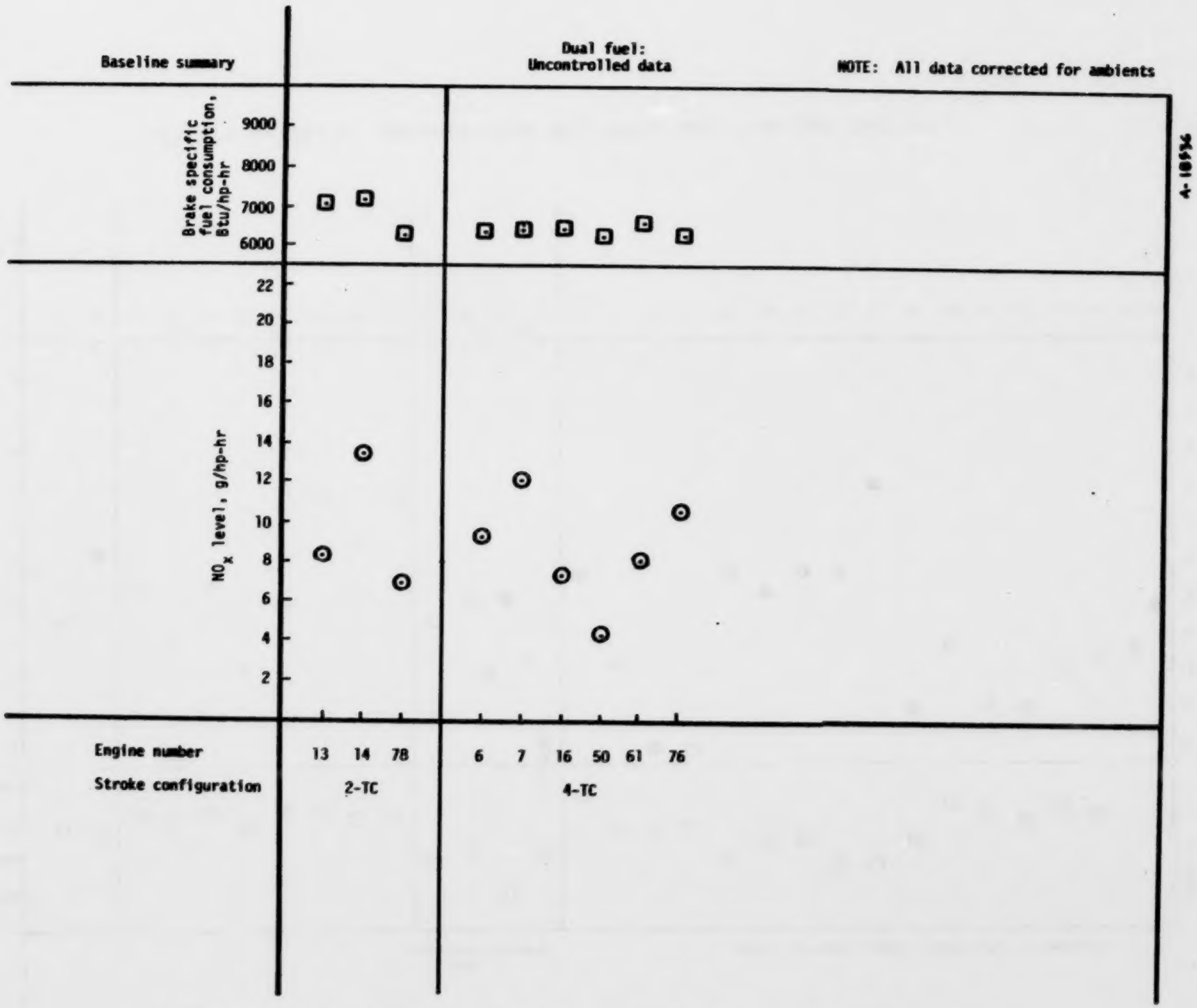


Figure 4-13(b). Uncontrolled $NO_x$ emissions from dual fuel engines.

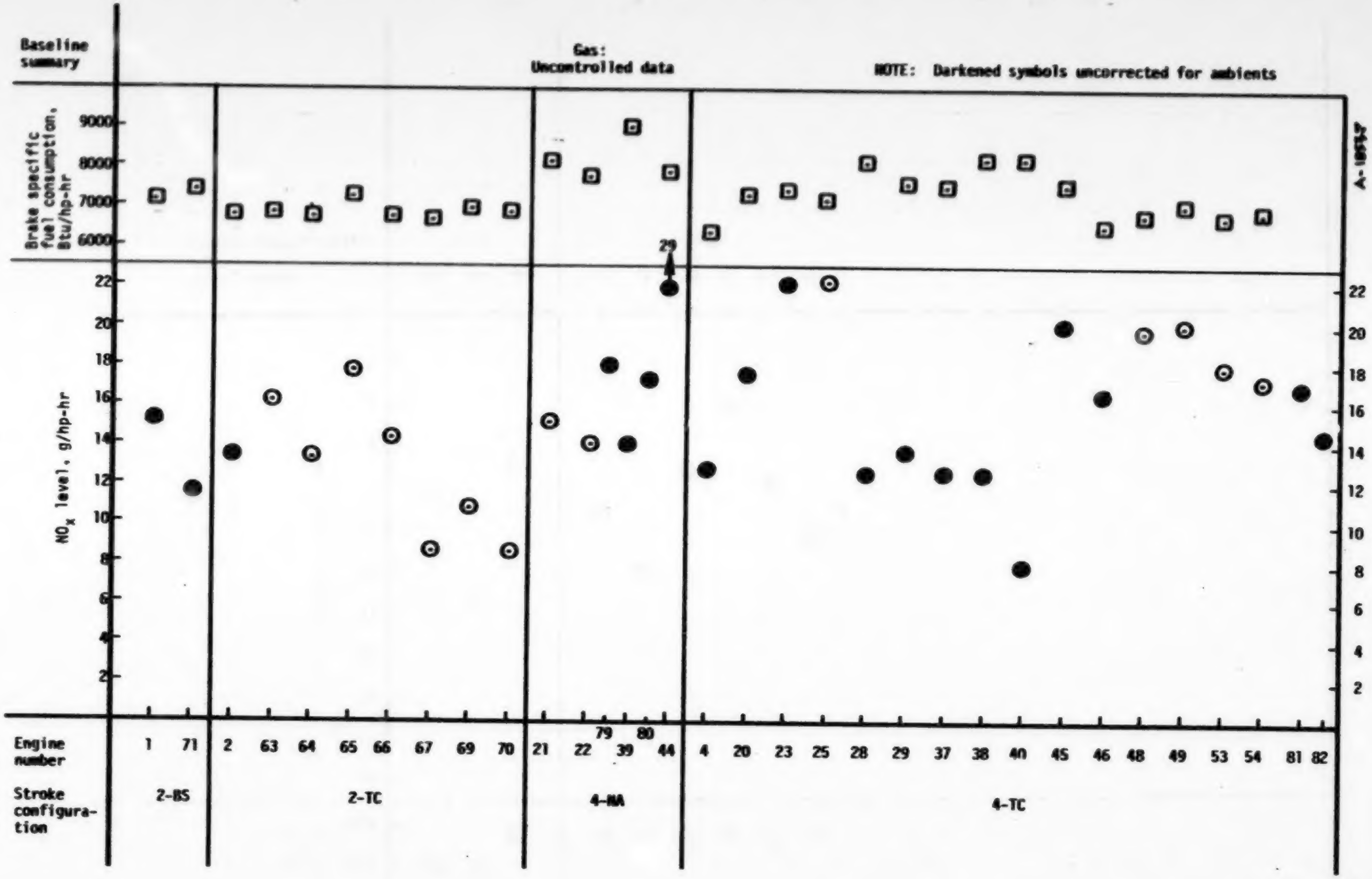


Figure 4-13(c). Uncontrolled NO$_x$ emissions from gas engines.

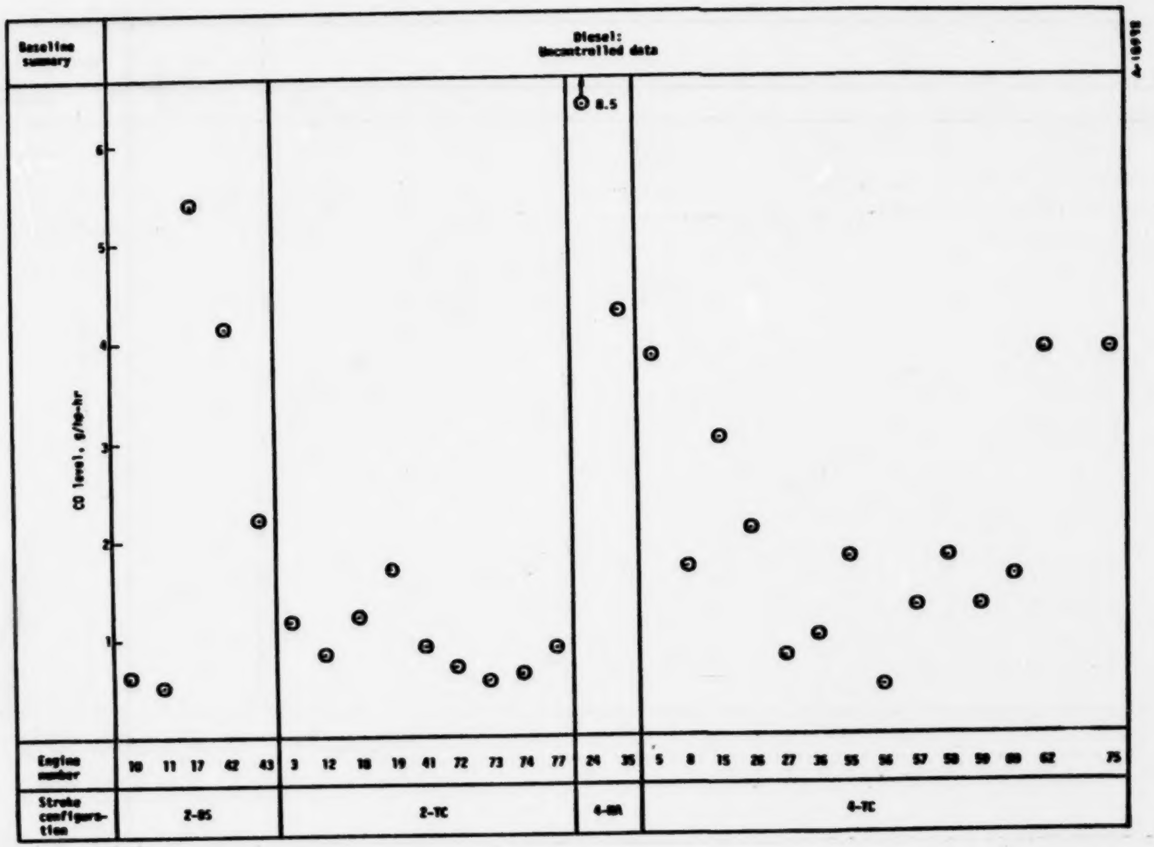


Figure 4-14(a). Uncontrolled CO emissions from diesel engines.

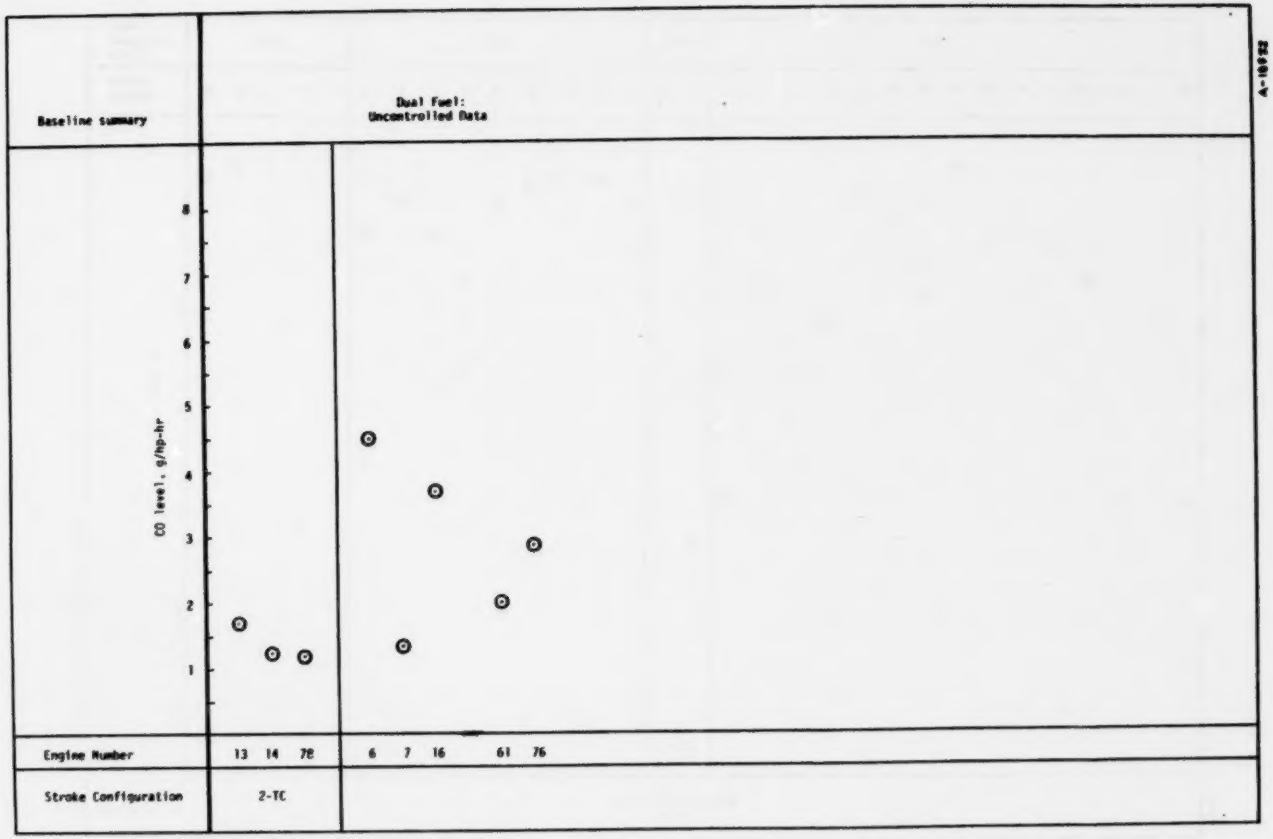


Figure 4-14(b). Uncontrolled CO emissions from dual fuel engines.

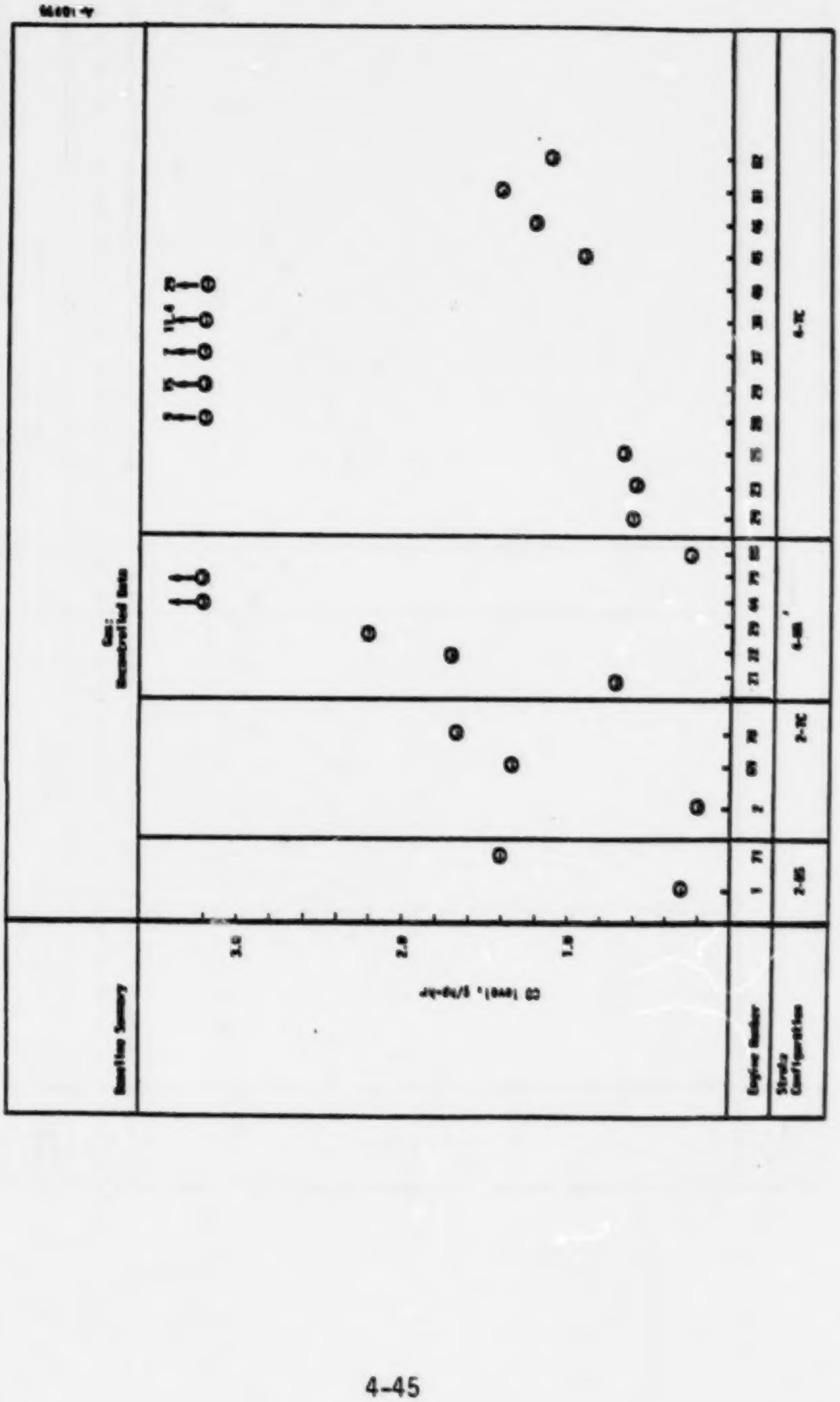
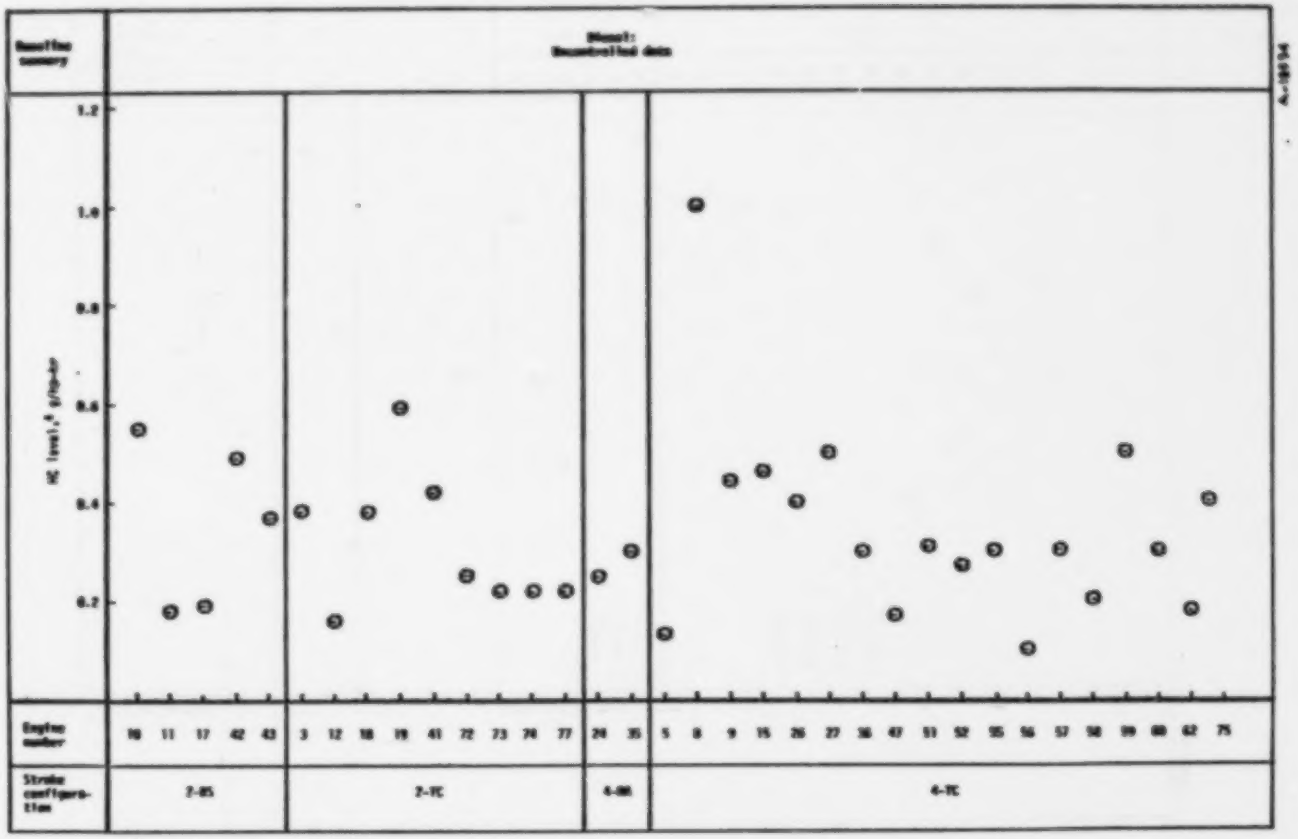


Figure 4-14(c). Uncontrolled CO emissions from gas engines.



aHC level assumed to be all nonmethane.

Figure 4-15(a). Uncontrolled HC emissions from diesel engines.

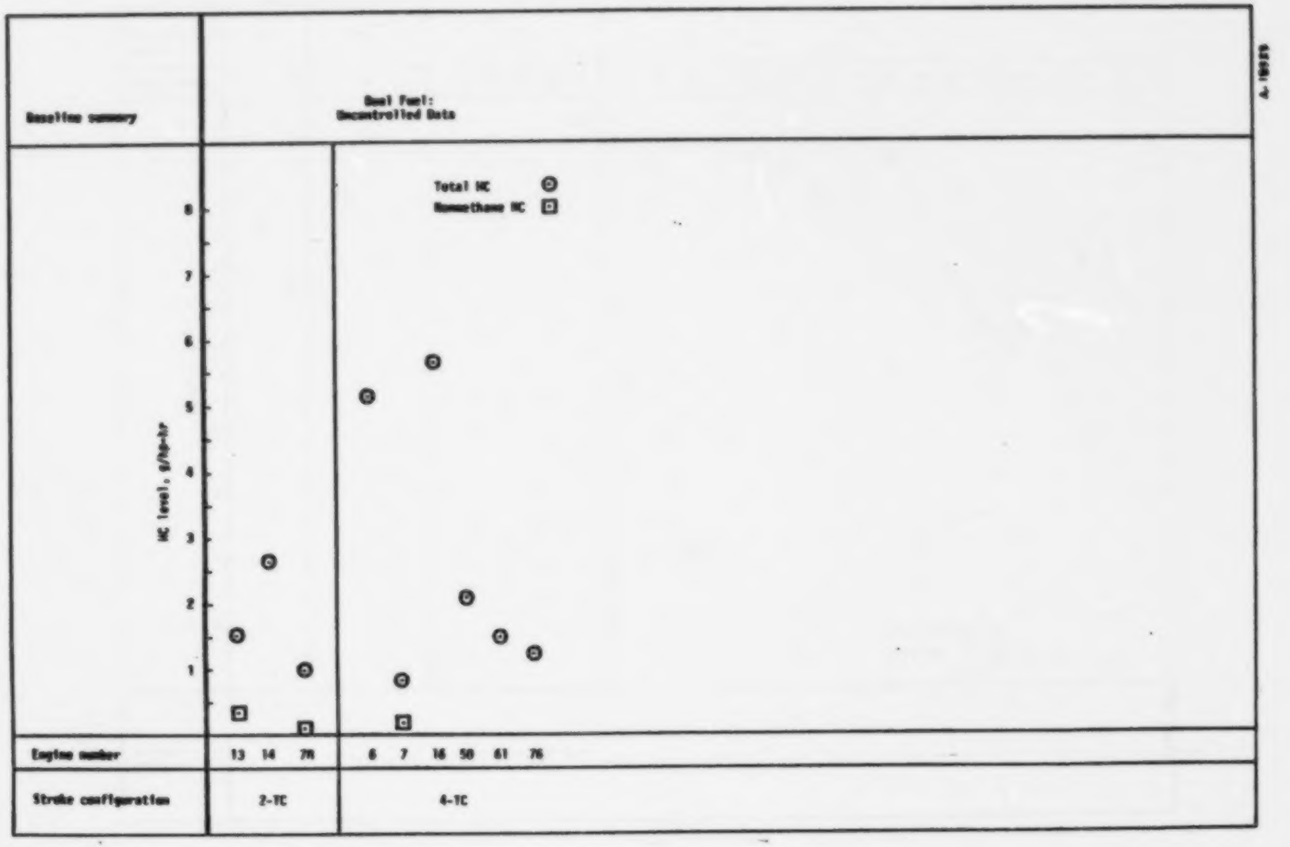


Figure 4-15(b). Uncontrolled HC engines from dual fuel engines.

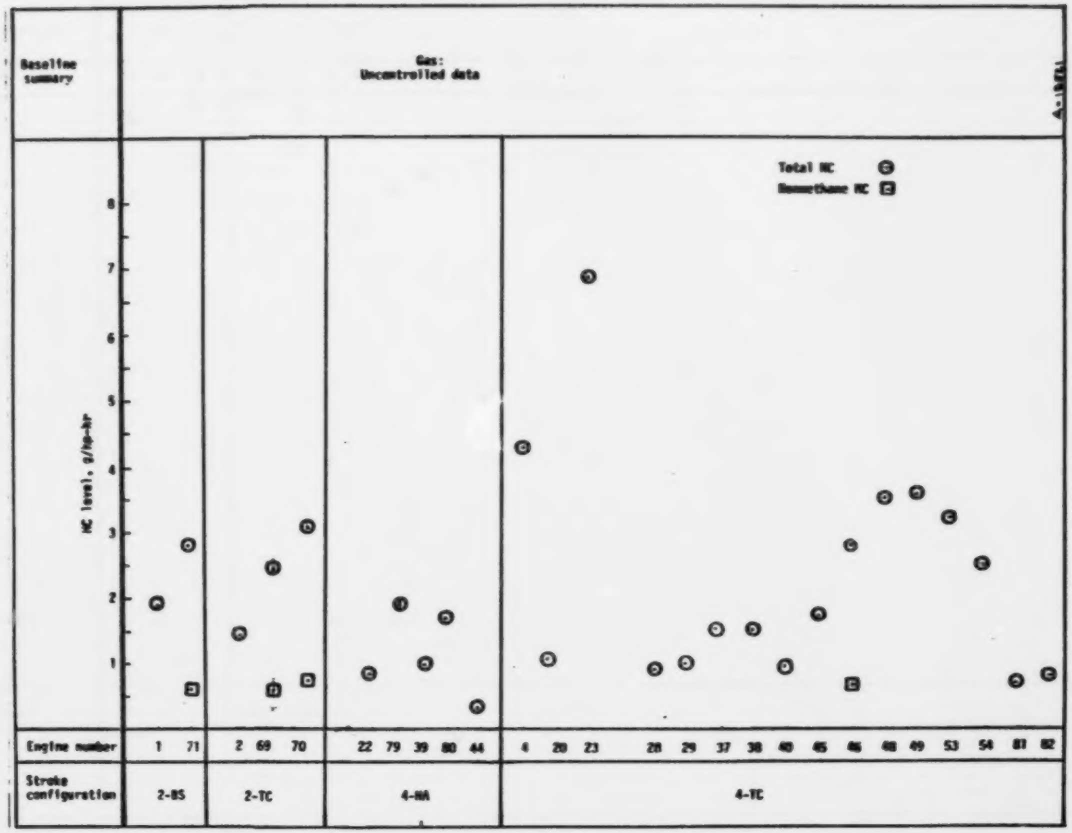


Figure 4-15(c). Uncontrolled HC emissions from gas engines.

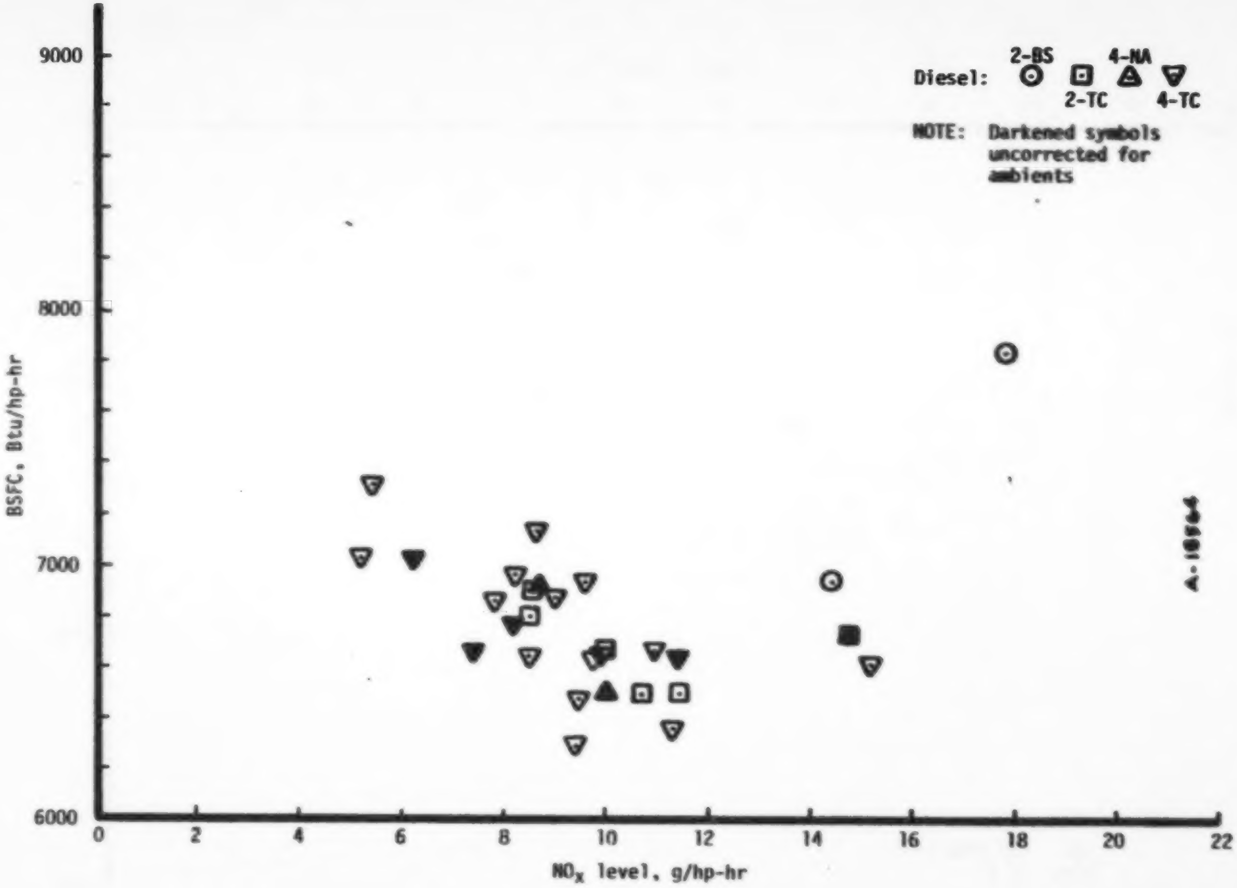


Figure 4-16(a). Uncontrolled $NO_x$ levels versus brake specific fuel consumption (BSFC) for diesel engines.

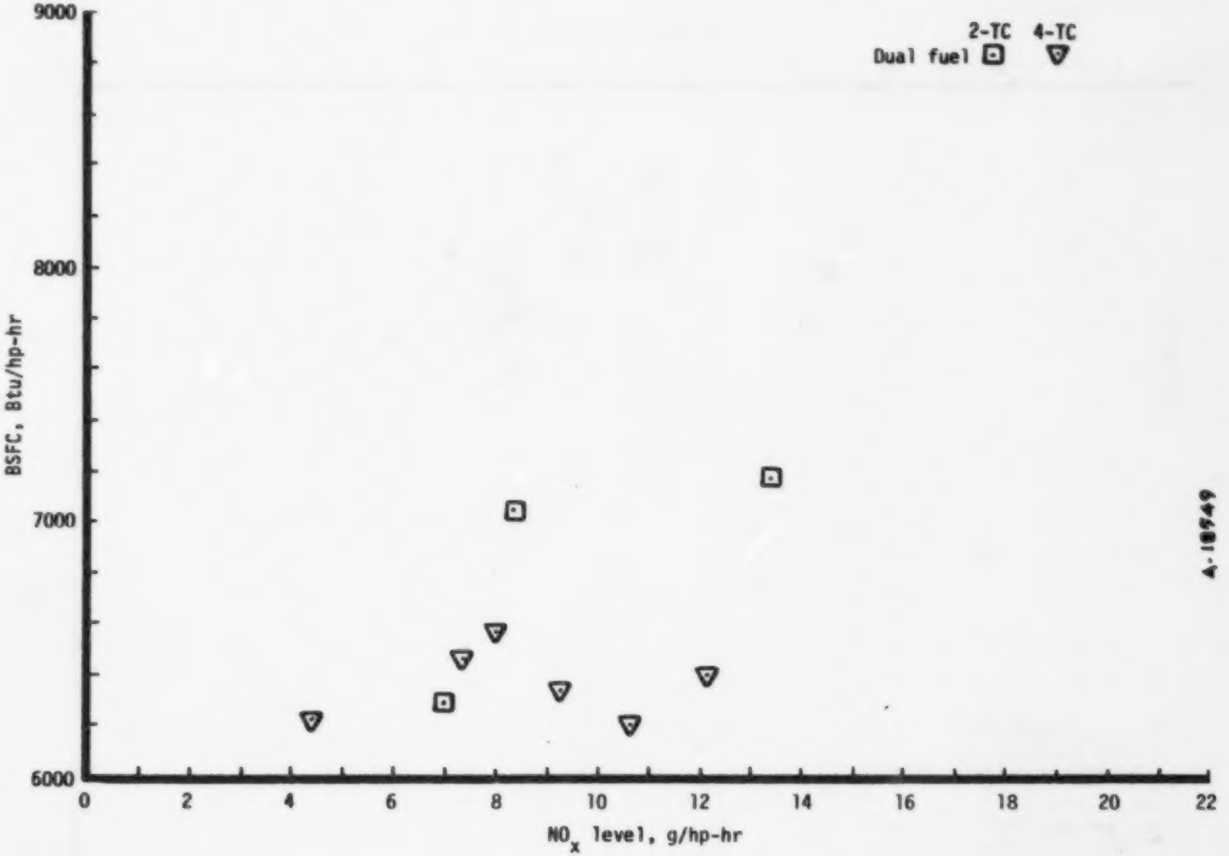


Figure 4-16(b). Uncontrolled $NO_x$ levels versus brake specific fuel consumption (BSFC) for dual fuel engines.

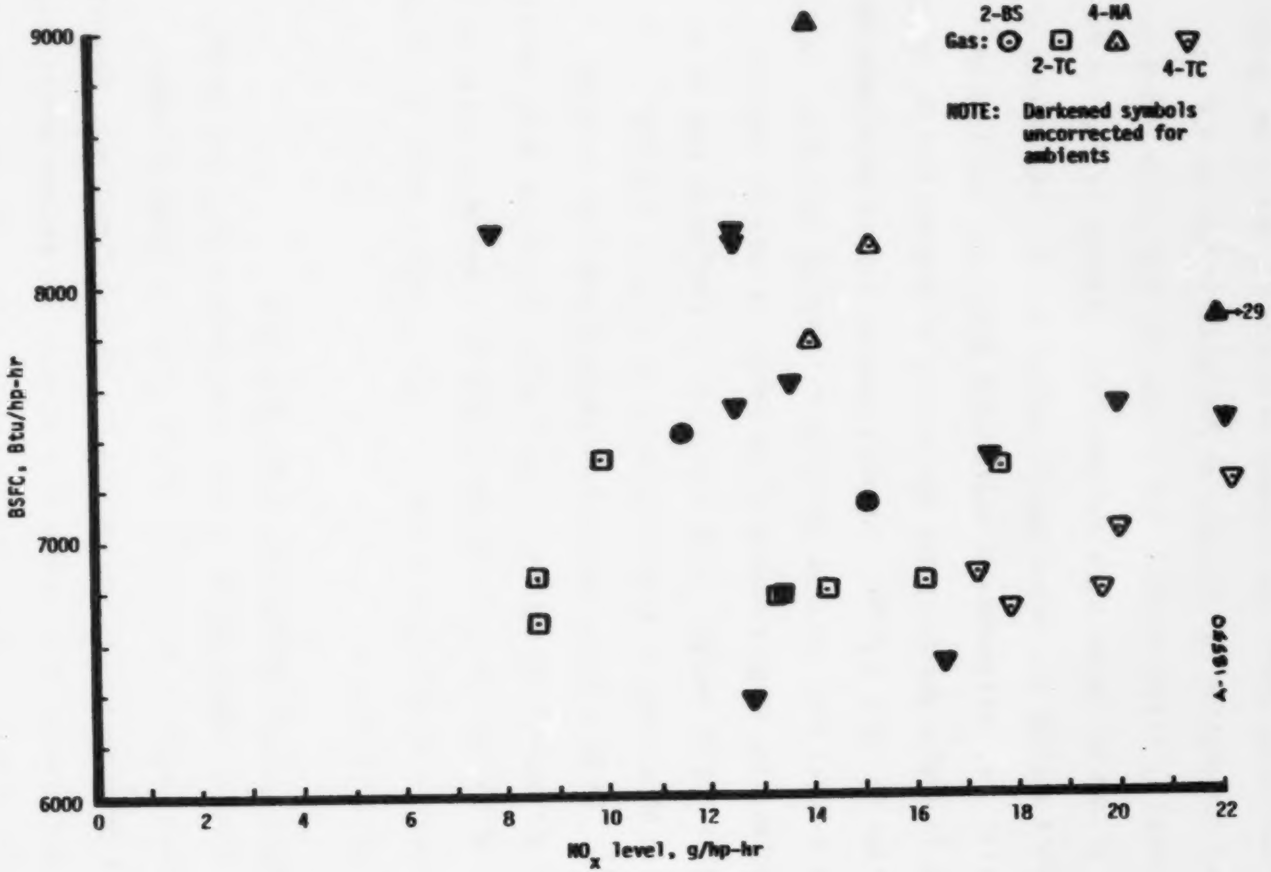


Figure 4-16(c). Uncontrolled $NO_x$ level versus brake specific fuel consumption (BSFC) for gas engines.

important in affecting uncontrolled $NO_x$ emissions and fuel consumption, changes in operating conditions intended to reduce $NO_x$ emissions, generally causing fuel consumption to increase, as discussed in Section 4.4.

As noted in Figure 4-13(a), not all the $NO_x$ data can be corrected for the effect of ambient temperature and humidity. (Ambient correction factors for large-bore engines are summarized in Section 4.2.1.) The effects of ambient variations, differences in measurement practices, and inherent differences in engine design on the variability of uncontrolled $NO_x$ emissions are summarized in Table 4-6(66). The data samples include only those data from Figure 4-13(a) that could be corrected for ambient variation. As this table indicates, the largest source of variations in data is inherent differences in engine design. This conclusion is similar to that of an investigation of sources of emission data variability in gasoline vehicles(67). In this study, researchers demonstrated that although measurement and ambient effects were significant, variations among vehicles caused most of the variations in $NO_x$ emissions for a series of tests on similar vehicles. Sources of emissions variability due to engine design are discussed in the following section.

### 4.3.3 Effect of Engine Variability on $NO_x$ Emissions

The emission data supplied by the manufacturers vary considerably. As discussed above, small variation can be attributed to using different measurements techniques or not correcting for ambient conditions. However, most of the differences in emissions from uncontrolled engines result from: (1) variations in the production of a particular model, (2) variations among different models of the same type (i.e., same strokes/cycle, air charging and fuel), or (3) variations in the number of cylinders for a given model. In

TABLE 4-6. SOURCES OF DATA VARIABILITY FOR UNCONTROLLED $NO_x$ EMISSIONS FROM LARGE-BORE ENGINES

| | Mean[a] | Std. Dev. | Sources of Variability[b] | | |
|---|---|---|---|---|---|
| | | | Ambient | Measurement | Engine |
| FUEL | $\overline{NO}_x$, g/hp-hr | $\sigma/\overline{NO}_x$, % | $\Delta A/\overline{NO}_x$, % | $\Delta M/\overline{NO}_x$, % | $\Delta E/\overline{NO}_x$, % |
| Diesel | 10.2 | 34 | 6 | High = 15<br>Low = 5 | 30 to 33 |
| Dual Fuel | 9.2 | 34 | 8 | High = 18<br>Low = 6 | 28 to 33 |
| Gas | 15.0 | 25 | 11 | High = 9<br>Low = 3 | 20 to 22 |

[a] Mean values were computed from the corrected data shown in Figures 4-13(a), (b), and (c).

[b] Sources of variability are related according to the law of error propagation (66), i.e.,

$$\sigma = \sqrt{\Delta A^2 + \Delta M^2 + \Delta E^2}$$

This approach assumes each uncertainty is independent of another, but there exists a statistical chance that the uncertainties could occur at the same time.

Ambient corrections are based on factors presented in Section 4.2.1

The estimate for measurement uncertainty includes a high and low value based on information presented in Section 4.2.2

Uncertainties due to engine design ($\Delta E$) were computed from $\sigma$, $\Delta A$, and $\Delta M$.

this section, the uncontrolled data base is evaluated for these sources of variability, and furthermore, the data are examined for trends related to differences in engine design, such as speed, torque (bmep), and manifold air temperature. All of these analyses use data corrected for ambient variations by the methods described in Section 4.2.1.

### 4.3.3.1 Production Variations

It is difficult to quantify variations in emissions among production units of the same model. Up to now, manufacturers have concentrated on obtaining emissions data for different engine models. Since no emission regulations (with the exception of smoke limits) have been in effect for stationary engines, there has been little impetus for manufacturers of large-bore stationary engines to make exhaust measurements of engines leaving the production line. (In general, fewer than 100 units are produced each year for stationary applications by any one manufacturer.) However, Colt and GMC/EMD, as well as numerous manufacturers of smaller bore, heavy duty engines for trucks have reported variations in emissions from production models.

One large volume manufacturer of medium-bore engines has shown that their laboratory units must emit at levels at least 25 percent lower than a performance standard, to insure that their production models will comply with the standard(68). This margin accounts for production variables that effect emission levels in mass produced engines. For 75 percent of this manufacturer's current engines meeting the Federal automotive emission standard of 16 g/hp-hr ($NO_x$ + HC), current variation in 1.34 g/hp-hr ($NO_x$ + HC). Moreover, this manufacturer believes that the magnitude of production variation is independent of the emission level, and this belief is shared by several other manufacturers of medium-bore engines(69). For large-bore engines, produced individually to higher tolerances, it is anticipated that this variation should be smaller.

Colt reported less than a 3-percent difference between production models in two $NO_x$ measurements of one in 1972, the other in 1975, from a two-stroke, blower-scavenged diesel engine(70). Such a small difference was unexpected. They suggest that the variation would more likely be of approximately 10 percent for production units, but they have no data to verify this estimate. Colt has measured $NO_x$ levels of production spark ignited engines (2-TC-G) within 3 percent of each other under similar ambient conditions(71).

GMC/EMD has reported average $NO_x$ levels and standard deviations for samples of their 2-TC and 2-BS diesel models(72). These results are summarized in Table 4-7(73). As this table suggests, these variations in $NO_x$ levels of production engines may have resulted from ambient variations. Inlet air temperatures varied over a wide range for both the turbocharged and blower-scavenged units, and humidity was not recorded. An attempt was made to determine whether these observed variations in emissions could be due to changing ambient conditions. First, a correction was computed for each extreme of the reported temperature range, using the methodology presented in Section 4.2.1. These two maximum variations were then compared to the reported data, to determine if temperature variations alone could account for the scatter. Next, a correction was computed for both the reported temperature variations and an assumed humidity variation ranging from 35 to 115 grains $H_2O$/lb dry air. These corrections were then compared with the production variability; the results are listed in Table 4-8.

TABLE 4-7. VARIATIONS IN $NO_x$ EMISSIONS FROM GMC/EMD PRODUCTION ENGINES (Reference 73)

| Type | # Cylinders | # Units | Variation (standard deviation as a % of average) | Ambient Temperature |
|---|---|---|---|---|
| 2-TC-D | 20 | 11 | ± 8 | 67-97°F |
| | 16 | 10 | ± 8 | |
| 2-BS-D | 16 | 13 | ± 6 | 63-128°F |
| | 12 | 8 | ± 5 | |

[a] Variations are computed for data that are uncorrected for ambient conditions.

## TABLE 4-8. POTENTIAL SCATTER IN $NO_x$ EMISSIONS FROM PRODUCTION ENGINES DUE TO AMBIENT HUMIDITY AND TEMPERATURE VARIATIONS

| Type | # Cylinders | Inlet Air Temperature Range | Scatter, Uncorrected for Ambient, % of Mean | Correction for Temperature Range[a], % of Mean | Correction for Temperature & Humidity Range[b], % of Mean |
|---|---|---|---|---|---|
| 2-TC-D | 20 | 67-97°F | ± 8 | +3 to -2<br>(67 to 97°F) | +16 to -11 |
| | 16 | | ± 8 | | |
| 2-BS-D | 16 | 63-128°F | ± 6 | +5 to -9<br>(63 to 128°F) | +15 to -14 |
| | 12 | | ± 5 | | |

[a] Using HD diesel ambient correction factor (see Section 4.2.1.3)
$K_{correction}$ = 1/(1 + B(T - 85)) where B = 0.0017 for 2-TC-D (based on 4-TC-D)
= 0.00235 for 2-BS-D
T = test temperature
(Reference temperature of 85°F)

[b] Using HD diesel ambient correction factor
$K_{correction}$ = 1/(1 + A(H - 75) + B(T - 75)) where A = -0.00231 for 2-TC-D (based on 4-TC-D)
= -0.00242 for 2-BS-D
H = test humidity ranging from 35 to 115 grains $H_2O$/lb dry air
(Reference humidity = 75 grains $H_2O$/lb dry air)

Based on these corrections, temperature variations alone could account for all of the variability in emissions from blower-scavenged engines, but not for turbocharged engines. If humidity were to vary over the range used for the calculations presented in Table 4-8, then differences in ambient temperatures and humidity could account for all the variability reported for these engines.

### 4.3.3.2 Model Variations

Variations in levels may be attributed to differences in models for a given manufacturer's engines. For example, $NO_x$ levels for a manufacturer's 4-TC-G models may vary due to differences in bore, stroke, turbocharger, configuration (inline cylinders vs. vee), compression ratio, aftercooler, and other engine design parameters. In an effort to identify the magnitude of model-to-model variations, average $NO_x$ levels and standard deviations were evaluated for different models of the same fuel type from each manufacturer.

Table 4-9 presents the results of this study. GMC/EMD, White Alco, and Colt are not included in this table since they each manufacture only one engine model (with different numbers of cylinders) per air charging method. Some of these models are configured for different fuels, for example, Colt markets their 38D8-1/8 opposed piston engine model as a gas, diesel, or dual fuel engine, either blower scavenged or turbocharged. GMC/EMD and White Alco manufacture one basic diesel-fueled, turbocharged design which differs primarily in number of cylinders and speed ratings. GMC/EMD also markets blower-scavenged units.

The other five manufacturers listed in Table 4-9 produce different engine models within a given engine type. $NO_x$ levels reported by Cooper for four 2-TC models varied by an average of 13 percent. These engines differed

TABLE 4-9. VARIATION IN $NO_x$ (g/hp-hr) DUE TO MODEL DIFFERENCES

| Manufacturer | Gas[a] | | | | | | | | Diesel[b] | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 4-TC | | | | 2-TC | | | | 4-TC | | | |
| | # | AVG | SD | % | # | AVG | SD | % | # | AVG | SD | % |
| Cooper | | | | | 4 | 14.6 | 1.8 | 13 | | | | |
| Delaval | 2 | 17.6 | 0.6 | 4 | | | | | 3 | 9.2 | 0.5 | 5 |
| Ingersoll-Rand | $3^c$ | 18.0 | 1.8 | 10 | | | | | | | | |
| White Superior | $2^c$ | 20.4 | 2.4 | 12 | | | | | | | | |
| Waukesha | $3^c$ | 12.8 | 0.6 | 4 | | | | | $2^c$ | 7.9 | 0.7 | 9 |

[a] Ambient correction for humidity only (see Table 4-2)

[b] Ambient correction for humidity and temperature (see Table 4-2)

[c] No ambient correction, ambients not recorded

SD — Standard deviation in g/hp-hr

% — SD/AVG

# — Number of models of same engine type

in bore, speed, number of cylinders, and torque (bmep), but were all operated at the same inlet and manifold air temperatures. Delaval's data indicated only a 4- to 5-percent variation between models for both gas- and diesel-fueled engines. The percent variations shown for Waukesha, Ingersoll-Rand, and White Superior, which were uncorrected for ambient conditions, should not be compared to the Cooper and Delaval results because differences due to ambient conditions could not be factored out. To the extent that conclusions can be drawn from such a small sample size, it appears that $NO_x$ emissions for any type of engine (given strokes/cycle, fuel, and air charging) vary more from manufacturer to manufacturer than among models in a manufacturer's line. Emission variations due to differences among manufacturers could be related to differences in speed, bmep, or manifold air temperature. This possibility is addressed in Section 4.3.3.4.

### 4.3.3.3 Variations With Number of Cylinders

Several manufacturers have suggested that $NO_x$ levels will vary for a basic engine design depending on the number of cylinders, since the manifold interacts with the turbocharger. Figure 4-17, which is a plot of $NO_x$ level (corrected for ambients) vs. number of cylinders, shows that $NO_x$ levels for 4-TC gas engines decrease significantly as number of cylinders increase, but $NO_x$ emissions from 4-TC diesel and dual-fuel engines do not indicate a trend with number of cylinders.

Figure 4-18 presents a different interpretation. The $NO_x$ levels (corrected for ambients) have been plotted vs. the number of cylinders for each manufacturer's engines, which tends to reduce other sources of emissions variation (such as design differences among manufacturers) that may have been

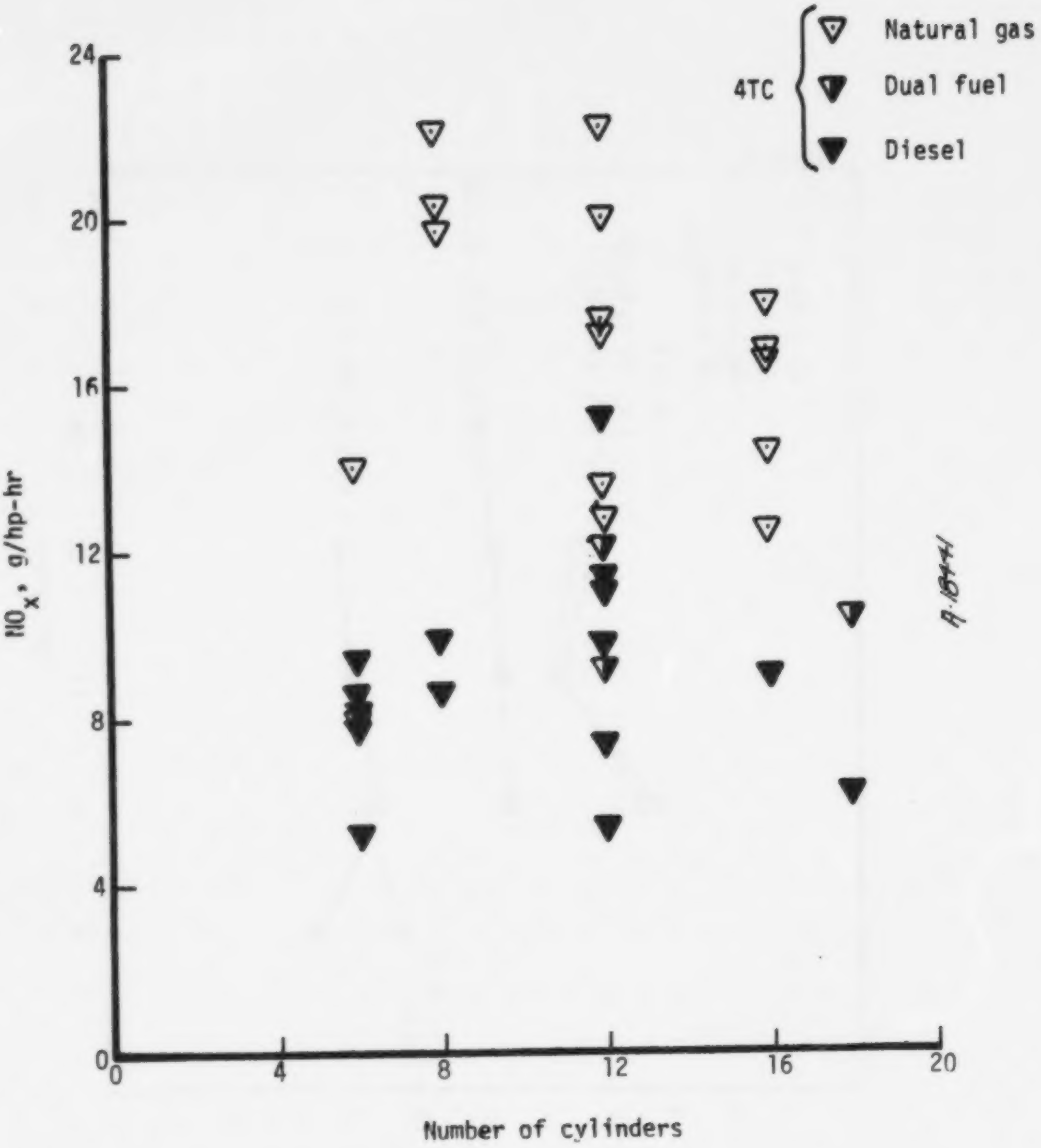


Figure 4-17. NO$_x$ production variation with number of cylinders.

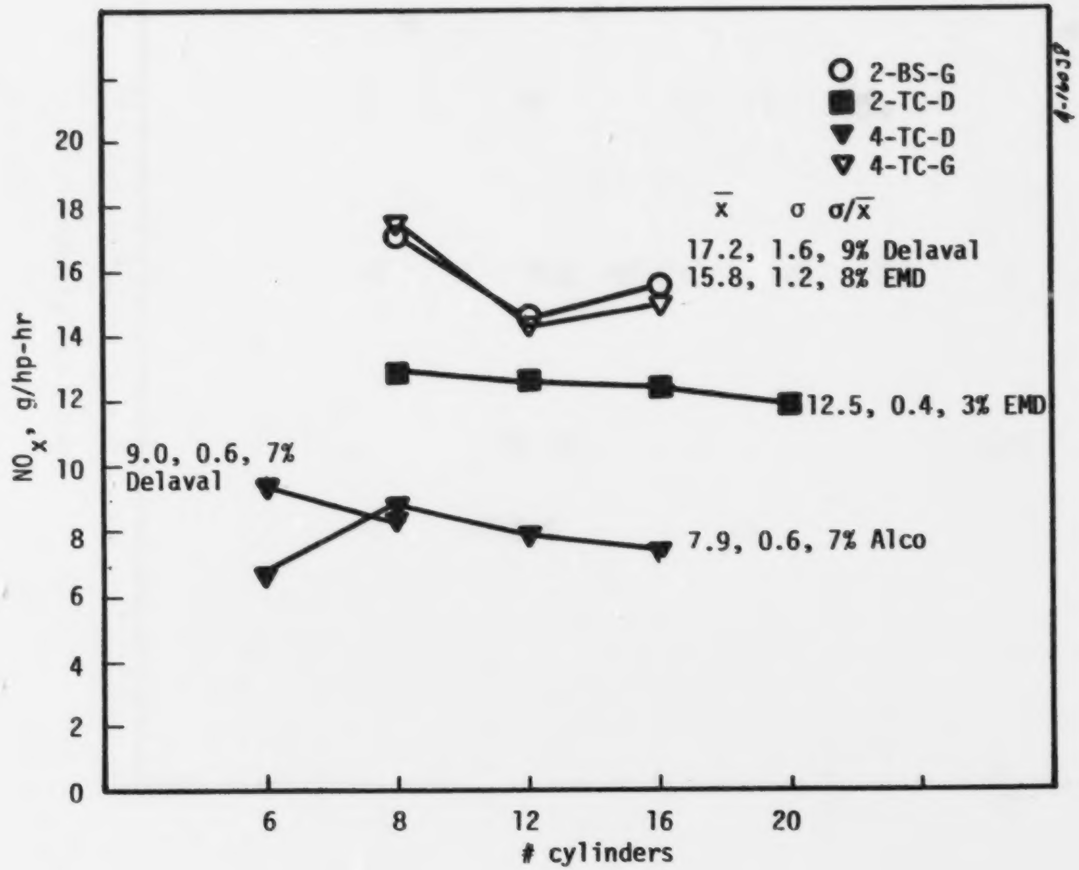


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Figure 4-18. Variation in $NO_x$ level with number of cylinders for individual engine models of different manufacturers.

reflected in Figure 4-17. Figure 4-18 indicates that there is no clear trend between $NO_x$ emissions and number of cylinders for either diesel or gas units. The effect of changing the number of cylinders causes uncontrolled $NO_x$ levels to vary from 3 to 9 percent. Because EMD data could not be corrected for ambients, this data may not represent the effect of differences in the number of cylinders.

### 4.3.3.4 Variations in $NO_x$ Level Due to Other Engine Variables

The results described above suggest that the variations in $NO_x$ levels reported for engines of a given type are probably due to design parameters that differentiate one manufacturer's engines from those of the others. Consequently, uncontrolled $NO_x$ data (corrected for ambients) were plotted vs. speed (rpm), manifold air temperature, and torque (bmep) to reveal any emission trends with these design parameters.

Figure 4-19 illustrates $NO_x$ level variation with speed for two engine types. The data for gas engines indicate increased $NO_x$ emission with in creased speed. This is in contradiction to what one would expect from the reasoning that decreased residence time (increased speed) should result in lowered $NO_x$ emissions. Apparently other factors (e.g., increased cylinder temperature or inherent design differences among different engines) are responsible for this trend. The 4-TC dual fuel and diesel $NO_x$ levels appear to decrease somewhat with increasing speed, as would be expected for lower exhaust gas chamber residence times.

Variations in $NO_x$ level with manifold air temperature are shown in Figure 4-20. The 4-TC-G $NO_x$ levels appear to be very sensitive to the design air manifold temperature, but the diesel and dual fuels $NO_x$ levels do not.

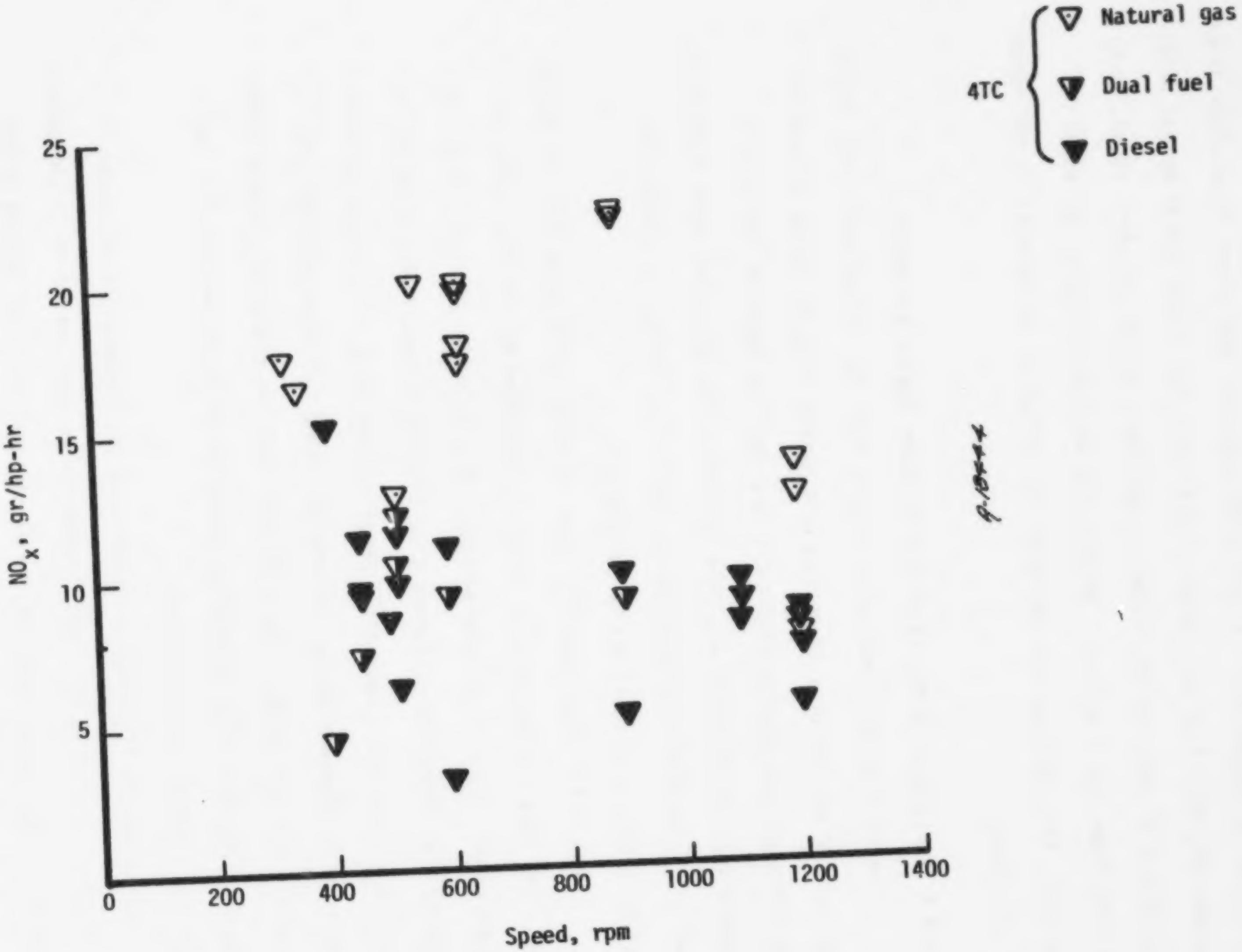

Figure 4-19. Variation in $NO_x$ level with speed.

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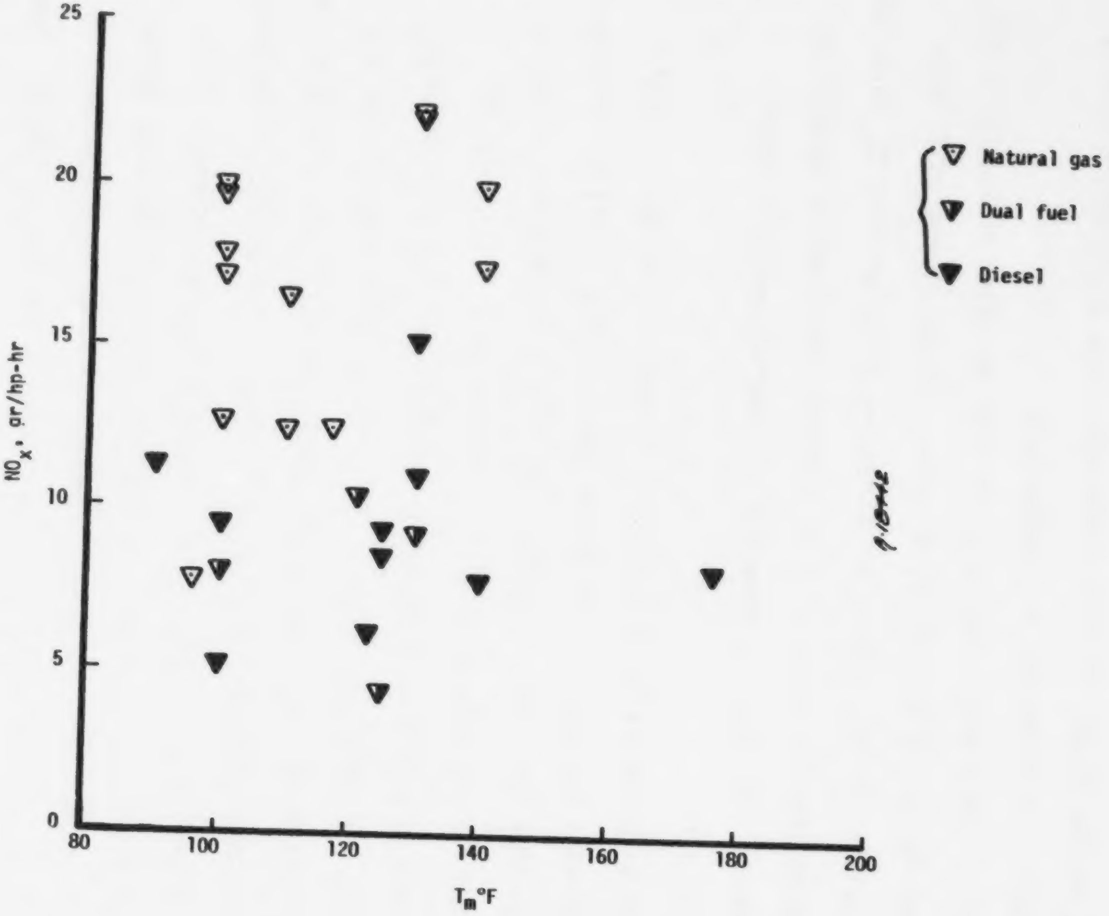

Figure 4-20. $NO_x$ variation with manifold temperature.

These results confirm expectations that $NO_x$ levels from premixed, vaporized fuel combustion in SI engines may be strongly influenced by the degree of aftercooling. On the other hand, CI engines are characterized by droplet combustion, and $NO_x$ production under these conditions would depend more on local A/F ratio than on overall air temperature.

Finally, Figure 4-21 illustrates the wide variation of $NO_x$ level with torque (bmep) for 4-TC-G and 4-TC-D, DF units. If the cluster of CI data around 150 psi is ignored, a trend of decreasing brake-specific $NO_x$ emissions with increasing bmep for these units is apparent. No trend is apparent for SI units, except that they are generally not manufactured with bmep's exceeding 200 psi.

Based on these preliminary studies, it appears that certain engine design parameters may explain more of the variation $NO_x$ levels for engines of a given type than variations in ambient humidity or temperature. That is, $NO_x$ emissions for any type engine (given strokes/cycle, fuel, and air charging) vary more from manufacturer to manufacturer than they do among models within a manufacturer's line. Differences among manufacturers are related to differences in speed, torque (BMEP), manifold air temperature, and combustion chamber design. For example, limited data show that $NO_x$ emissions from 4-TC diesel and dual fuel engines decrease as speed increases. $NO_x$ emissions from 4-TC, natural gas (SI) engines increase directly as design manifold air temperature increases. However, no clear trends can be established for the effects of number of cylinders and torque (bmep). These observations suggest that some form of weighted average is required to characterize uncontrolled $NO_x$ emissions from each of the three fuels. This approach is discussed in the following sections.

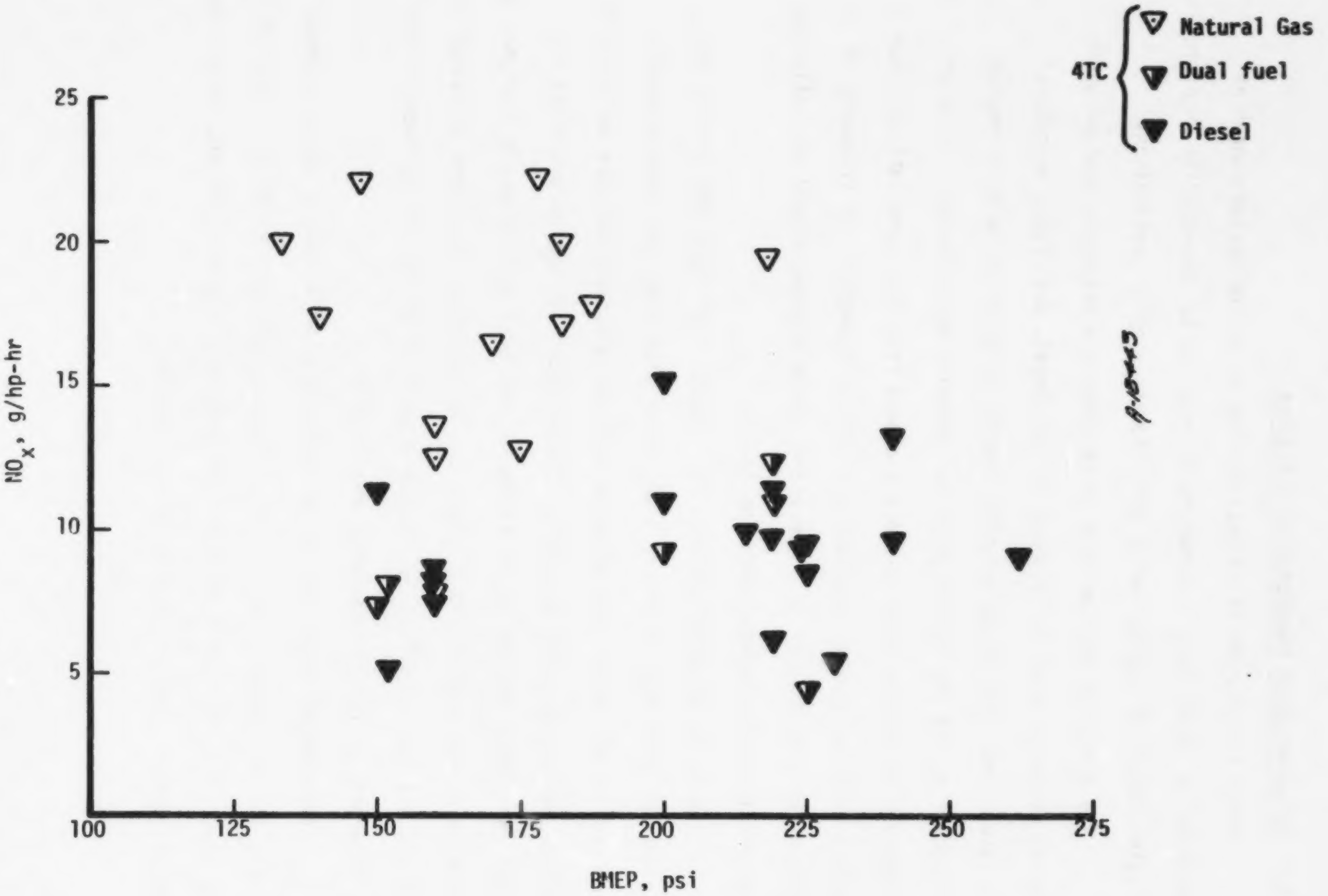

Figure 4-21. $NO_x$ variation with torque (bmep).

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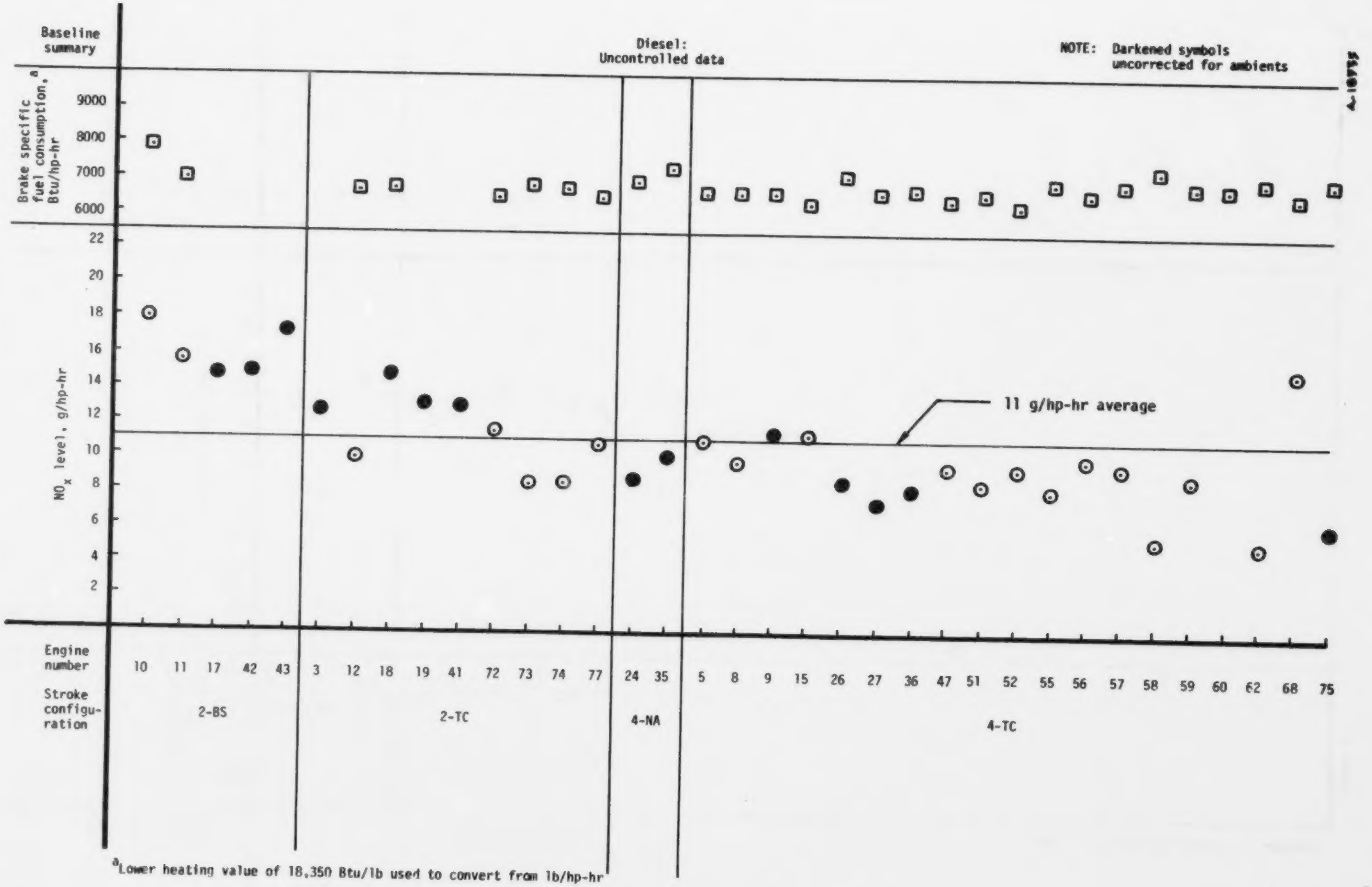
### 4.3.4 Sales-Weighted Uncontrolled Emissions

Since the sources of variability due to engine design cannot be specifically identified, a procedure is required to characterize uncontrolled emission levels of engines which are sold for similar applications.

The procedure adopted here is to compute a weighted, average uncontrolled emission level for engines in the diesel, dual fuel, or natural gas categories. The three weighted levels are based on sales of engine horsepower during the past 5 years for domestic applications. Sales of horsepower to standby services were excluded from this computation, since engines sold for standby applications will be exempted from standards of performance (see Chapter 9). Therefore, these engines should not influence the selection of regulated emission levels.

The sales-weighted averages for diesel, dual fuel, and natural gas engines are presented in Figure 4-22, which also show each manufacturer's uncontrolled $NO_x$ data. The weighted averages are based on data corrected for ambient conditions where possible. The weighted average uncontrolled $NO_x$ level for diesel engines is 11.0 g/hp-hr, for dual fuel units, 8.1 g/hphr, and for natural gas engines, 15.0 g/hp-hr. The emission reductions discussed in Section 4.4 and summarized in Chapter 6 can then be applied to these levels to determine potential regulated levels of $NO_x$.

Measurement uncertainties are associated with each of these weighted levels and are shown in Table 4-10. These uncertainties should be applied to the controlled $NO_x$ levels that are determined by applying the $NO_x$ reductions demonstrated by the alternative control systems.



[a] Lower heating value of 18,350 Btu/lb used to convert from lb/hp-hr

Figure 4-22(a). Sales-weighted uncontrolled $NO_x$ emissions for diesel engines.

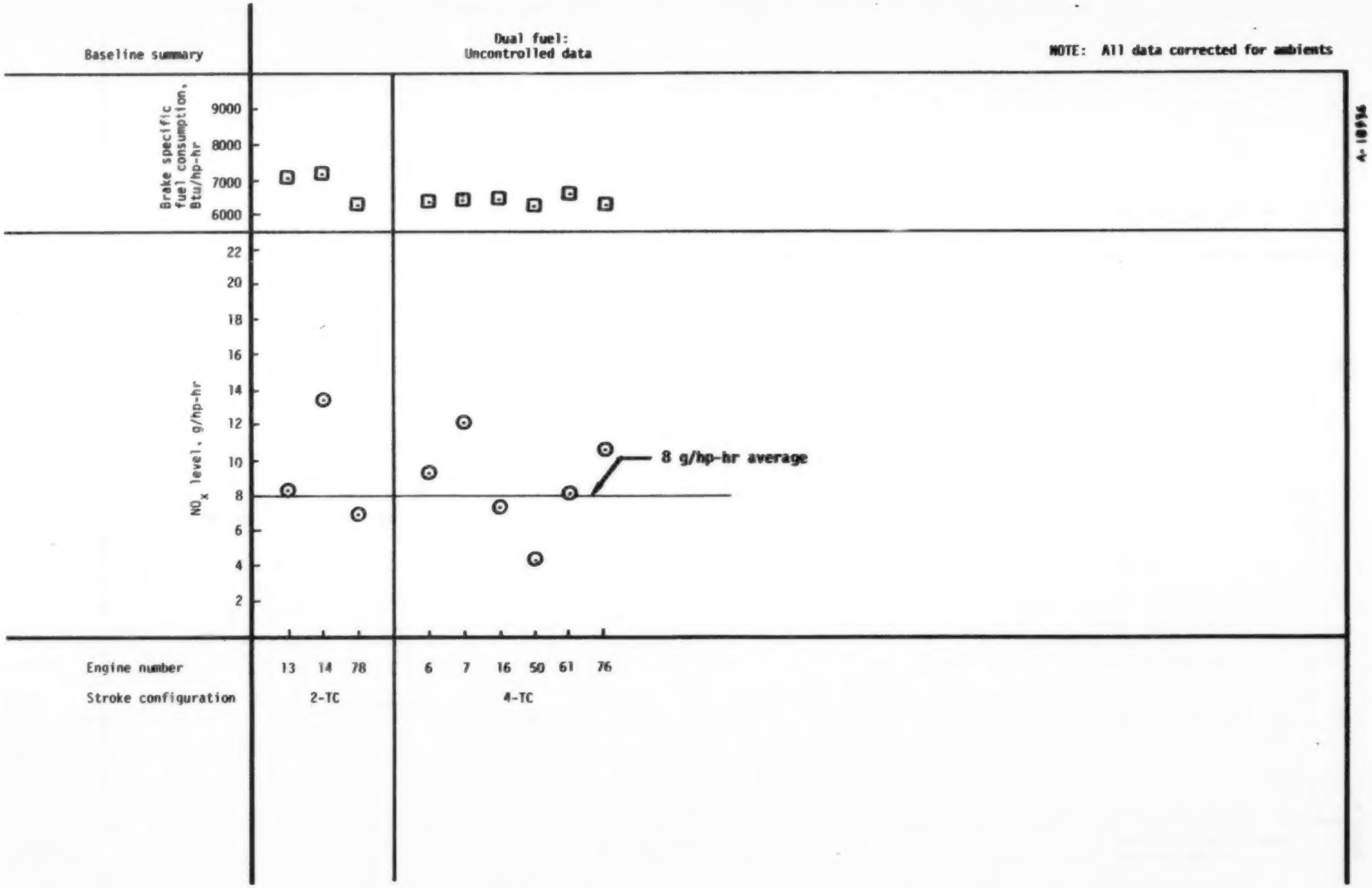


Figure 4-22(b). Sales-weighted uncontrolled $NO_x$ emissions for dual fuel engines.

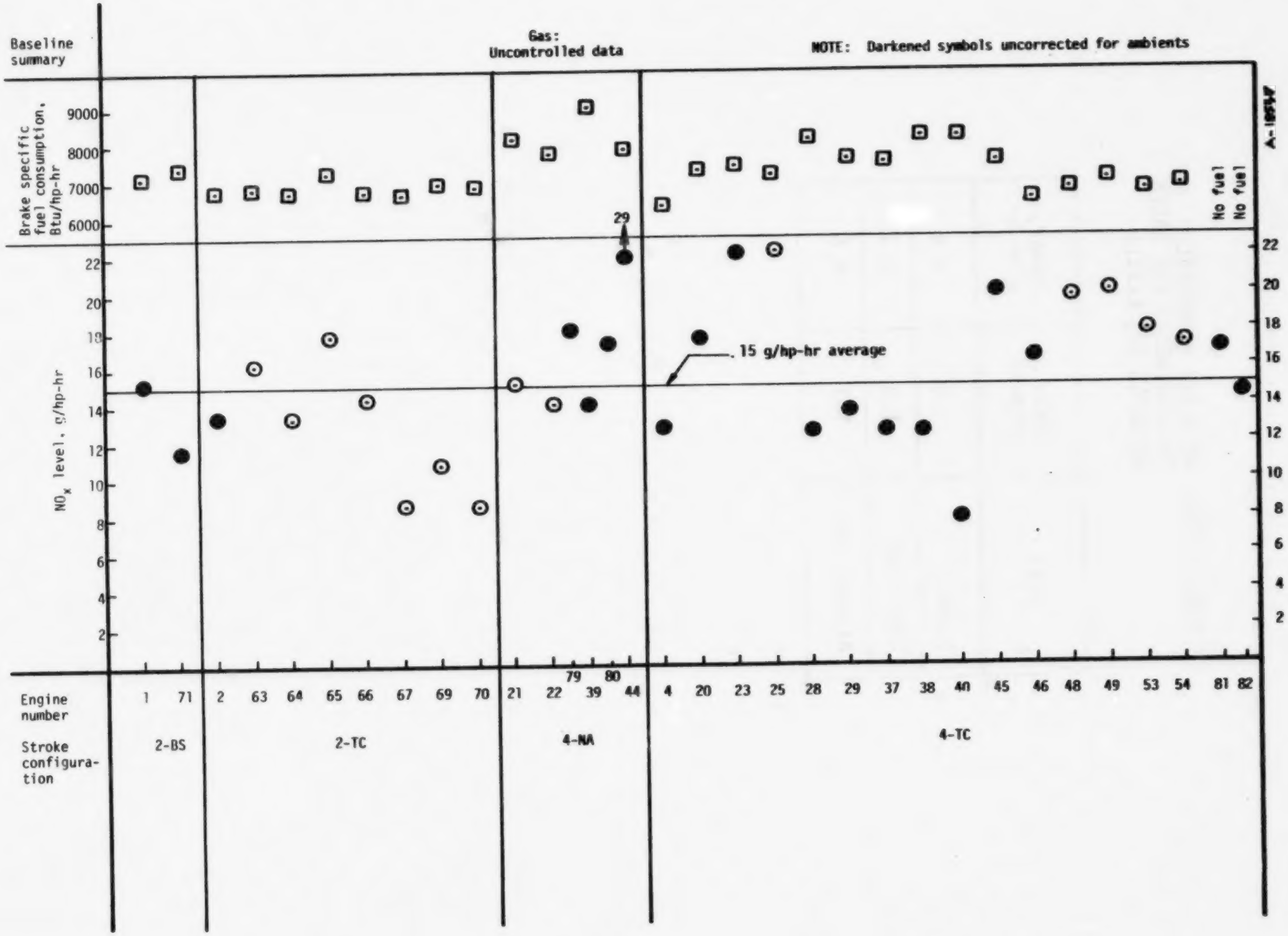


Figure 4-22(c). Sales-weighted uncontrolled $NO_x$ emissions for gas engines.

TABLE 4-10. WEIGHTED MEASUREMENT UNCERTAINTIES FOR SALES WEIGHTED $NO_x$ LEVELS

| Fuel | Upper, g/hp-hr | Lower, g/hp-hr |
|---|---|---|
| Diesel | 1.0 | 0.9 |
| Dual Fuel | 1.0 | 0.4 |
| Natural Gas | 1.3 | 0.5 |

## 4.4 $NO_x$ EMISSION REDUCTION TECHNIQUES

This section describes techniques that have been used, or are being evaluated for use, to control $NO_x$ emissions from IC engines. Sections 4.5 and 4.6 discuss control techniques that are designed primarily to reduce other pollutants (HC, CO, smoke). Since $SO_x$ emissions are directly related to the fuel sulfur content, these emissions are discussed in 4.4.13, combustion of nonstandard fuels.

The data presented here come from tests on engines whose operating conditions were altered or which were equipped with emission reduction devices. These tests were conducted in manufacturers' laboratories rather than in field installations. The discussion of each potential control technique centers on how the technique works, its effectiveness, resulting fuel penalties, effects on other pollutant emissions, technical limitations to its applications, and cost implications (i.e., additional fuel, maintenance, or hardware expense incurred by the application of the control).

Most techniques for controlling emissions from IC engines involve engine modifications rather than add-on tail gas treatment facilities. Engines are designed for optimum operation within one or more of the following constraints: application, initial cost, fuel consumption, maintenance requirements, reliability, and commitment of a company's engineering staff to a design.[5/] Each engine design satisfies the constraint

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[5/] Stationary reciprocating IC engines, and particularly the large ones, may be required to deliver thousands of hours of continuous operation at rated load under varying ambient conditions without significant maintenance, or to start without failure by remote control and deliver full power within 10 seconds. Since these are severe demands, manufacturers feel committed to a proven design and are, therefore, reluctant to make significant design changes (e.g., changed piston or cylinder shape or strokes per cycle).

differently. For example, one engine may operate at 4° BTDC while another at 5° BTDC to meet the same $NO_x$ emission level. Therefore, the data are grouped by engine type and fuel in the tables and graphs that follow. In addition, whenever there is a specific, known reason why one type of engine responds differently to the application of controls than does another, these differences are explained in the accompanying discussion of the control technique.

The reductions in $NO_x$ shown here were achieved by investigators for current production engines. In general, no attempt was made to optimize the engine for the controlled settings (i.e., decrease fuel consumption, reduce maintenance, etc.). Thus, these results must be viewed as those achievable if no attempt is made to reoptimize an engine's controlled setting.

As discussed in Section 4.2.2, the manufacturers' data were measured using one of four measurement practices (EPA, DEMA, SAE, EMD). Although differences in three of these practices relative to EPA's may cause uncertainties in the reported levels, the data are considered adequate for the purpose of setting standards of performance since these are small in comparison to those in emissions due to inherent differences in engine design. (Measurement uncertainties for uncontrolled emissions are discussed in Sections 4.3.2 and 4.3.4.) Furthermore, the reported emissions data have been corrected to standard conditions of humidity and temperature (when ambient data were recorded) using the ambient correction factors presented in Section 4.2.1 Dashed lines on the figures in this section indicate $NO_x$ reductions after ambient correction.

The control systems discussed in this section are listed below in their order of presentation.

1. Derating (D)
2. Retard (R)

3. Changed air-to-fuel ratio (A/F)
4. Turbocharging with aftercooler (TC)
5. Reduced manifold air temperature (MAT or M)
6. Exhaust gas recirculation (EGR) -- internal (IE) and external (EE)
7. Water induction ($H_2O$)
8. Combustion chamber redesign (CCR)
9. Catalytic converters
10. Combinations of the above

Several abbreviations will be used on the charts and tables in this chapter; they are listed in Table 4-11. Fuel consumption data on the charts and tables are based on a lower heating value (LHV) of 18320 Btu/lb (10160 kcal/kg) for No. 2 diesel oil.

A qualitative summary, by pollutant, of the effect of each control technique on each engine type as shown by the available data is presented in Table 4-12. Sections 4.4.1 through 4.4.10 give quantitative results for each of these techniques. Graphs are presented that show (1) the $NO_x$ reduction for the largest degree of control applied for each manufacturer and (2) the effect on emissions and fuel consumption as the amount of control is varied. The information presented in the second set of graphs has been normalized by the baseline or uncontrolled level. This condition is denoted on the graphs with a subscript "U" for uncontrolled. The controlled condition is denoted with a subscript "C" for the controlled level. Section 4.4.11 summarizes the data presented for each of the above control approaches.

Then in Section 4.4.12, the effect of $NO_x$ control on the emission of other pollutants is examined. This review will help to illustrate whether standards of performance may be required for other pollutants in addition to

## TABLE 4-11. ABBREVIATIONS FOR ENGINE TYPE AND EMISSION CONTROL TECHNOLOGY

| Abbreviation | Explanation |
|---|---|
| Fuel | |
| D | Diesel |
| DF | Dual Fuel |
| G | Gas (i.e., natural gas) |
| Strokes/Cycle | |
| 2 | 2-stroke/cycle engine |
| 4 | 4-stroke/cycle engine |
| Air Charging | |
| BS | Blower scavenged |
| NA | Naturally aspirated |
| TC | Turbocharged (and intercooled) |
| Control Technology[a] | |
| D | Derating |
| R | Retard |
| TC | Turbocharged (and intercooled) |
| A or A/F | Increased air-to-fuel ratio |
| M or MAT | Decreased inlet manifold air temperature |
| EGR(I) | Exhaust gas recirculation — internal |
| EGR(E) | Exhaust gas recirculation — external |
| S[b] | Increased speed |
| INJ | Modified injectors |
| $H_2O$ | Water induction |
| W/F or w/f | Water-to-fuel mass induction ratio |
| PC | Precombustion chamber |
| VT | Variable throat precombustion chamber |
| CCR | Combustion chamber redesign |
| Cat | Catalytic converter |

[a] RM and RMA are used to denote the combined use of retard, decreased air temperature, and increased air-to-fuel ratio

[b] Increased speed included because data are available, but it is not considered to be a viable control technique (Subsection 5.3.12)

TABLE 4-12(a). EFFECTS OF CONTROLS ON ENGINES LARGER THAN 350 $IN^3/CYL$: $NO_X$ EMISSIONS

| Fuel | Diesel | | | | Dual Fuel | | | | Natural Gas | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Strokes/Cycle | Two | | Four | | Two | | Four | | Two | | Four | |
| Control / Air Charging | BS | TC | NA | TC | BS | TC | NA | TC | BS | TC | NA | TC |
| Derate | ↓ | | | ↓ | | | | ↓ | ↓ | ↓ | ↓ | ↓ |
| Retard | ↓ | ↓ | | ↓ | | ↓ | | ↓ | ↓ | ↓ | ↓ | ↓ |
| A/F | | | | ↓ | | ↓ | | ↓ | ↓ | ↓ | ↓ | ↓ |
| TC | ↓ | — | | — | | — | | — | ↓ | — | ↑ | — |
| MAT | ↓ | ↑ | | ↓ | | ↓ | | ↓ | ↓ | ↓ | | ↓ |
| INJ | ↓ | ↑↓ | | | | | | | | | | |
| EGR(I) | | | | ↑ | | | | ↑ | ↓ | | ↓ | |
| EGR(E) | | ↓ | | | | ↑ | | | | | | |
| $H_2O$ | ↓ | | | ↓ | | | | ↓ | | | ↓ | ↓ |
| CR | | | ↓ | ↓ | | | | | | | | |

↑ Denotes emission increase with application of control
↓ Denotes emission decrease with application of control
↑↓ Denotes conflicting data with application of control
— Denotes no change in emissions with application of control
Blank indicates no data available on effect

TABLE 4-12(b). EFFECTS OF CONTROLS ON ENGINES LARGER THAN 350 $IN^3$/CYL: CO EMISSIONS

| Fuel | Diesel | | | | Dual Fuel | | | | Natural Gas | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Strokes/Cycle | Two | | Four | | Two | | Four | | Two | | Four | |
| Control / Air Charging | BS | TC | NA | TC | BS | TC | NA | TC | BS | TC | NA | TC |
| Derate | ↑ | | | ↓ | | | | ↑↓ | ↑ | ↑ | ↑ | ↑ |
| Retard | ↑ | ↑ | | ↑ | | ↑ | | ↑ | ↑ | ↑ | | ↓ |
| A/F | | | | ↑↓ | | ↑ | | ↓ | | ↑ | — | |
| TC | ↑↓ | | | | | | | | ↓ | | ↓ | |
| MAT | ↓ | ↓ | | ↑↓ | | ↑ | | ↑↓ | ↑ | ↑ | | ↑ |
| INJ | ↓ | ↑↓ | | | | | | | | | | |
| EGR(I) | | | | ↓ | | | | ↓ | ↑ | | — | |
| EGR(E) | | ↑ | | | | ↑ | | | | | | |
| $H_2O$ | ↓ | | | ↑↓ | | | | ↓ | | | ↑ | ↑ |
| CR | | | | | | | | | | | | |

↑ Denotes emission increase with application of control
↓ Denotes emission decrease with application of control
↑↓ Denotes conflicting data with application of control
— Denotes no change in emissions with application of control
Blank indicates no data available on effect

TABLE 4-12(c). EFFECTS OF CONTROLS ON ENGINES LARGER THAN 350 $IN^3/CYL$: HC EMISSIONS

| Fuel | Diesel | | | | Dual Fuel | | | | Natural Gas | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Strokes/Cycle | Two | | Four | | Two | | Four | | Two | | Four | |
| Control \ Air Charging | BS | TC | NA | TC | BS | TC | NA | TC | BS | TC | NA | TC |
| Derate | | | | ↑ | | | | ↑ | ↑ | ↑ | ↑ | ↑ |
| Retard | — | ↑↓ | | ↑↓ | | ↑ | | ↓ | ↑ | ↑ | ↑ | ↑ |
| A/F | | | | ↑↓ | | ↑ | | ↑ | | ↑ | | ↑ |
| TC | ↓ | | | | | | | | ↓ | | | |
| MAT | ↑ | ↑↓ | | ↑↓ | | ↑ | | ↓↑ | ↑ | ↑ | | ↑ |
| INJ | ↓ | ↓ | ↓ | | | | | | | | | |
| EGR(I) | | | | ↓ | | | | ↑ | ↑ | | | |
| EGR(E) | | — | | | | ↓ | | | | | | |
| $H_2O$ | ↑ | | | ↑↓ | | | | ↑ | | | ↑ | ↑ |
| CR | | | | | | | | | | | | |

↑ Denotes emission increase with application of control
↓ Denotes emission decrease with application of control
↑↓ Denotes conflicting data with application of control
— Denotes no change in emissions with application of control
Blank indicates no data available on effect

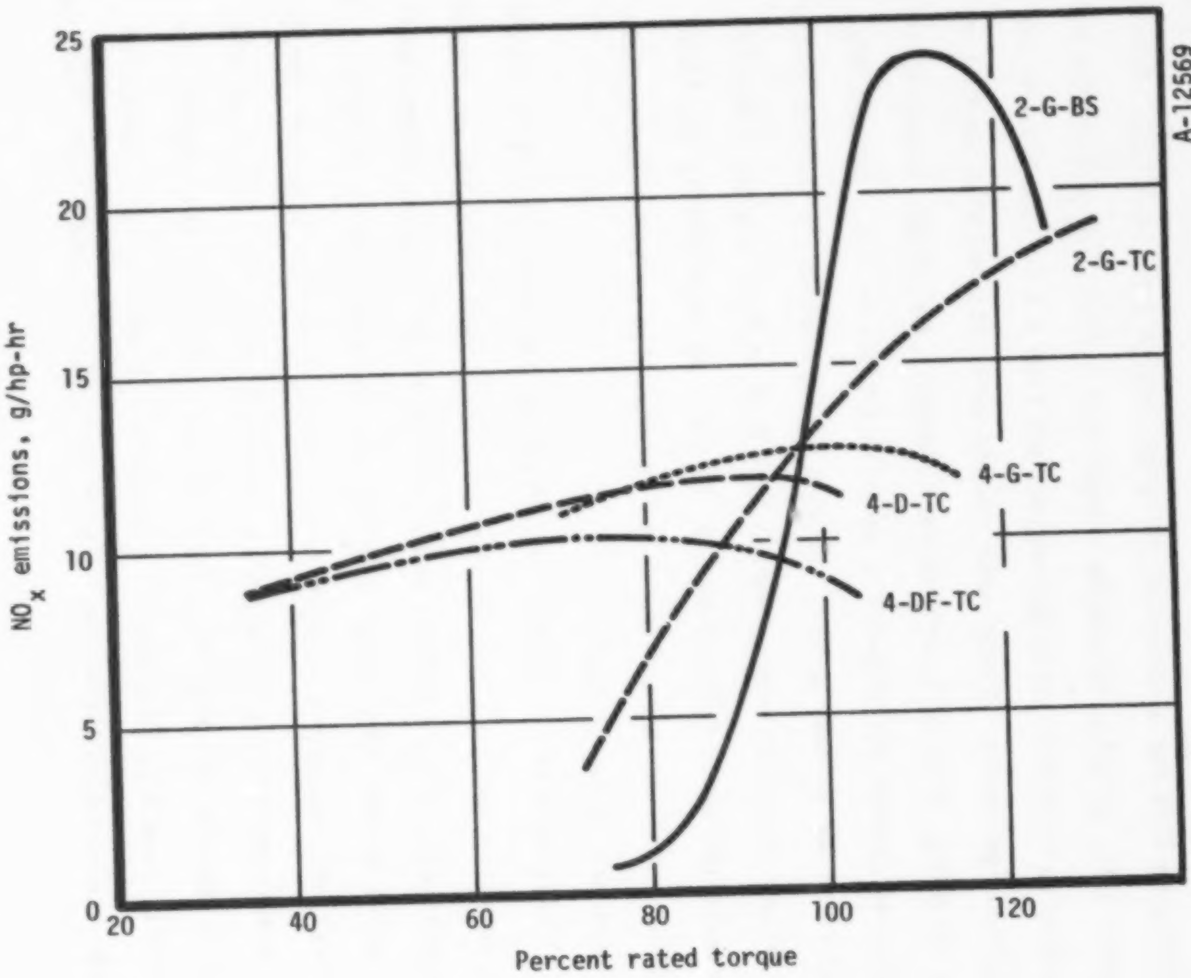
$NO_x$. Finally, Section 4.4.13 discusses the effect of burning nonstandard fuels on emissions from stationary IC engines.

The data that appear in this chapter can also be found in Appendix C.1, where they are tabulated by engine. This appendix presents all the available data on emissions and fuel consumption for large-bore engines.

### 4.4.1 Derating

When a manufacturer advertises or sells an engine, he guarantees that it will produce a given power at a stated speed. These conditions are called "rated conditions" and can be specified either for maximum, intermittent, or continuous operating conditions. The maximum rating usually refers to the peak power that can be achieved by the engine, but manufacturers generally recommend that the engine not be operated at this level. Intermittent ratings typically indicate the power output that the engine can produce for a 1-hour period with at least a 1-hour period of operation at, or below, the continuous rating before the next surge to the intermittent level(74). Continuous rating, of course, applies to uninterrupted operation (e.g., 24 hours per day, 365 days per year with shutdowns for maintenance only).

An engine can be derated by restricting its operation to a lower level of power production than normal for the given application. The effect of derating is to reduce cylinder pressures and temperatures and thus to lower $NO_x$ formation rates. Although $NO_x$ exhaust concentrations (i.e., moles of $NO_x$ per mole of exhaust) are reduced, it is quite possible for this reduction to be no greater than the power decrease. In such a case brake specific emissions (i.e., grams $NO_x$ per horsepower-hour) are not reduced. This is especially true for four-stroke turbocharged engines as shown in Figure 4-23(75). In addition, air to fuel ratios change less with derating for



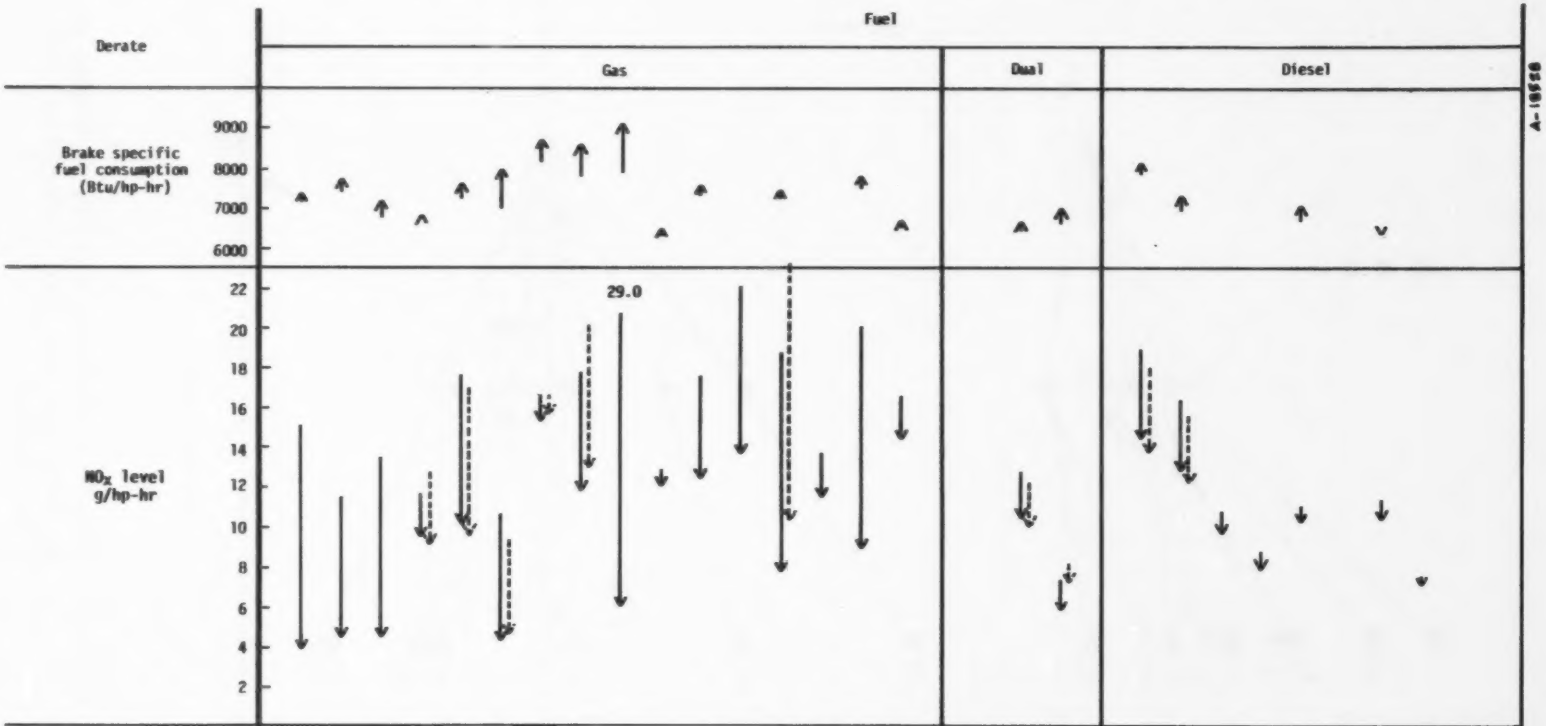
Note: All engines from same manufacturer

Figure 4-23. $NO_x$ emissions versus torque at constant speed (Reference 75).

turbocharged engines than for naturally aspirated or blower-scavenged units. Thus $NO_x$ emissions are less responsive to derating for turbocharged engines. Derating also reduces the engine's operating temperature, which then results in higher CO and HC emissions. This happens because the temperature dependent reactions that reduce these pollutants are less active(76).

Demonstrated $NO_x$ emission reduction levels due to derating are shown in Figure 4-24 for a number of different engine types and fuel. Based on these data, emission reductions ranged from 1.2 to 23.0 g/hp-hr for naturally aspirated or blower-scavenged engines and from 0.2 to 10.8 g/hp-hr for turbocharged units. Since these results were obtained with varying amounts of derating, it is more informative to compare the effectiveness of this emission control technique on a normalized basis -- i.e., percent $NO_x$ reduction per percent derate. On this basis, results for naturally aspirated or blower-scavenged engines varied from 0.25 to 6.2, whereas those for turbocharged units varied from 0.01 to 2.6. No relationship was found between normalized effectiveness and uncontrolled emission level, number of strokes per cycle, or fuel.

Figure 4-25 illustrates the effect of different amounts of derating on $NO_x$ emissions and fuel consumption for diesel, dual fuel, and gas engines, respectively. Figure 4-25(a) shows that derating decreases brake specific $NO_x$ emission from some diesel engines, but increases them from others. In general, $NO_x$ reductions range from 2 to 25 percent for 25 percent derating and brake specific fuel consumption increases range from 2 to 5 percent. With 50-percent derating, $NO_x$ reductions range from 15 to 45 percent and fuel penalties from 4 to 16 percent. Engines No. 10 and 11, both two-stroke blower-scavenged, achieved the largest $NO_x$ reduction (and highest fuel penalties).



| Derate | Gas | | | | | | | | | | | | | | | | Dual | | Diesel | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Percent derate | 12 | 20 | 25 | 13 | 20 | 25 | 25 | 25 | 25 | 14 | 25 | 25 | 25 | 25 | 25 | 25 | 18 | 25 | 25 | 5 | 25 | 25 | 38 | 27 | 25 |
| Normalized reduction[a] | 6.2 | 3.0 | 2.6 | 1.4 | 2.1 | 2.3 | 0.3 | 1.3 | 0.4 | 0.3 | 1.1 | 1.5 | 2.3 | 0.6 | 2.2 | 0.5 | 0.9 | 0.8 | 0.9 | 4.2 | 0.3 | 0.5 | 0.2 | 0.3 | 0.2 |
| Uncontrolled fuel consumption | 7127 | 7400 | 6775 | 6760 | 7265 | 6942 | 8130 | 7760 | 7840 | 6374 | 7300 | 7440 | 7220 | 7590 | 7500 | 6500 | 6400 | 6570 | 7844 | 6951 | 7175 | 6930 | 6677 | 6366 | 6667 |
| Percent increase | 2.6 | 4.1 | 6.1 | 0.3 | 4.3 | 14.4 | 6.4 | 9.6 | 15.3 | 0.8 | 3.0 | 1.7 | 2.8 | -- | 3.5 | 2.0 | 3.1 | 6.1 | 2.6 | 4.1 | -- | -5.5 | 4.8 | -0.7 | -- |
| Air charging | BS | BS | TC | TC | TC | TC | NA | NA | NA | TC | TC | TC | TC | TC | TC | TC | TC | TC | BS | BS | TC | NA | TC | TC | TC |
| Strokes/cycle | 2 | 2 | 2 | 2 | 2 | 2 | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 2 | 2 | 2 | 4 | 4 | 4 | 4 |
| Engine number | 1 | 71 | 2 | 64 | 65 | 69 | 21 | 22 | 44 | 4 | 20 | 23* | 25 | 29 | 45 | 46 | 7 | 61 | 10 | 11 | 12 | 24* | 5 | 15 | 27 |

[a] Percent $NO_x$ reduction/percent derate.

[b] Speed reduction, const. torque.

Figure 4-24. Effect of derate on $NO_x$ emissions.

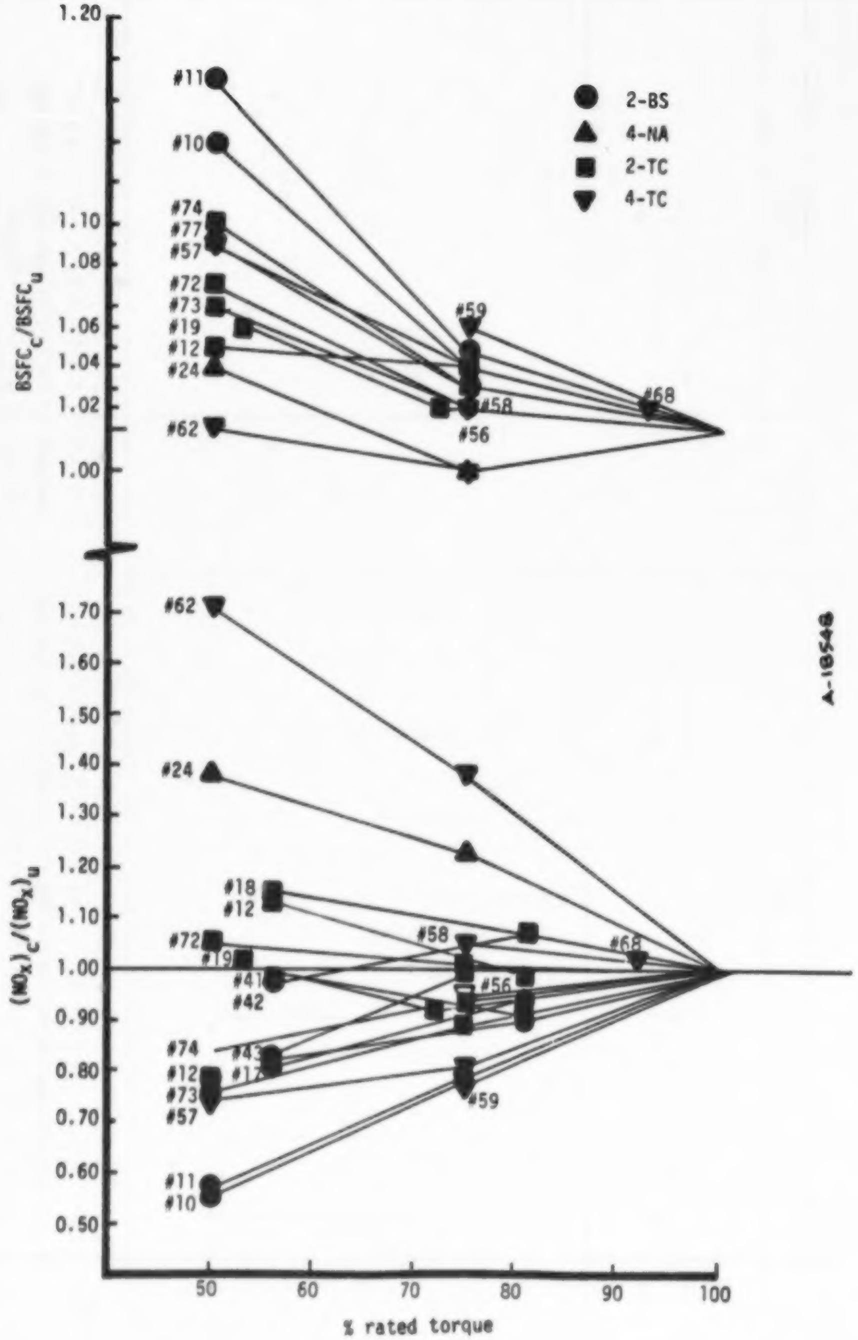


Figure 4-25(a). Effect of different amounts of derate on $NO_x$ emissions and fuel consumption from diesel engines.

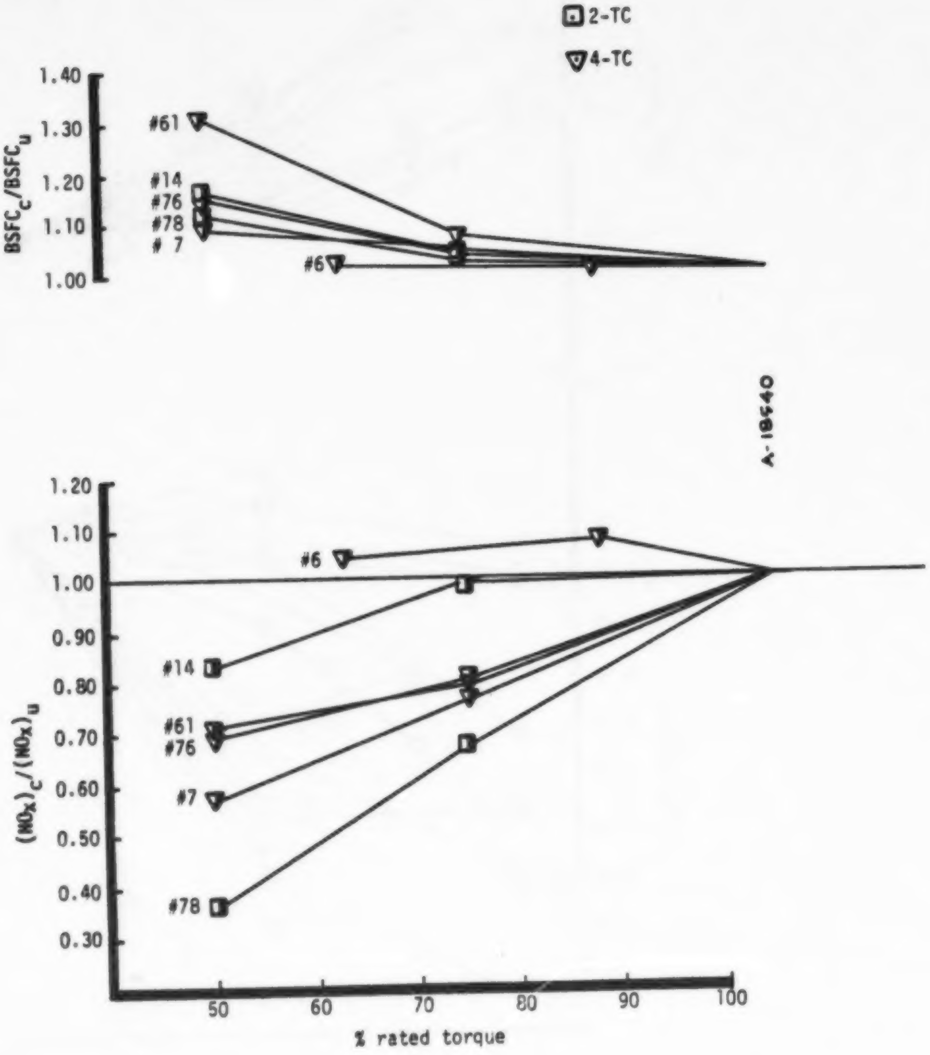


Figure 4-25(b). Effect of different amounts of derate on $NO_x$ emissions and fuel consumption on dual fuel engines.

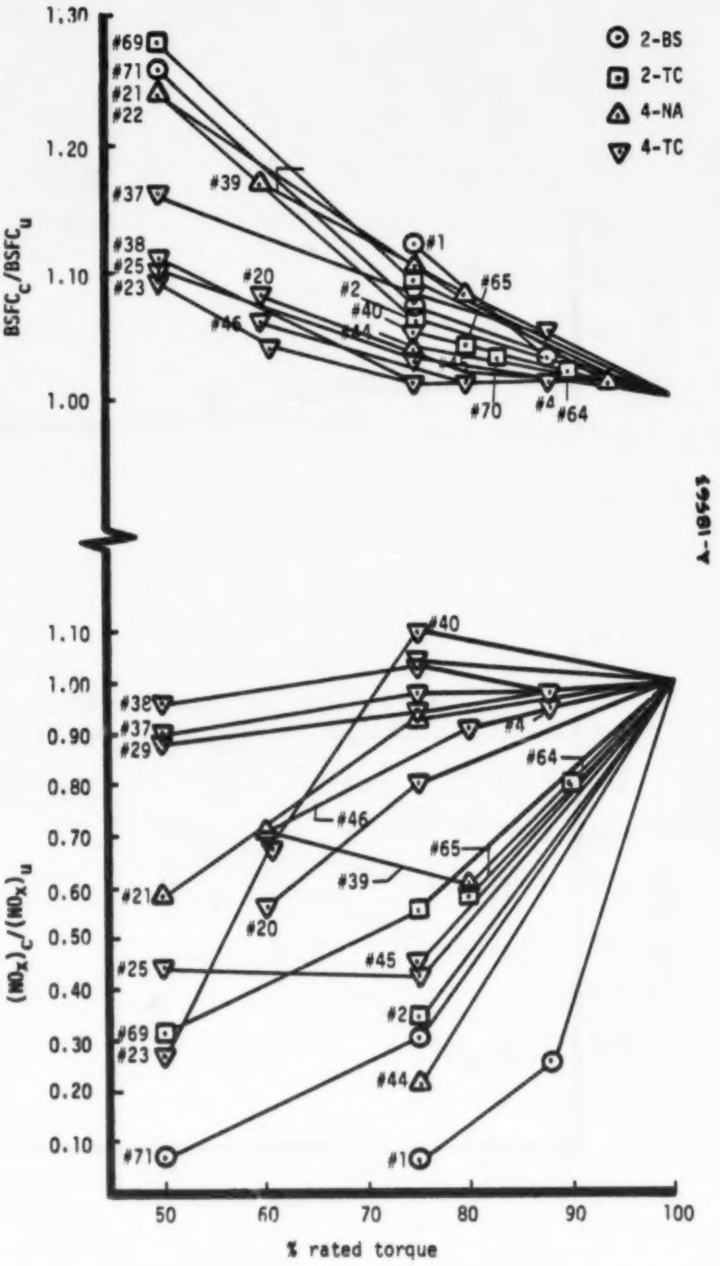


Figure 4-25(c). Effect of different amounts of derate on $NO_x$ emissions and fuel consumption for gas engines.

Figure 4-25(b) also shows mixed results for derating dual-fuel engines but with substantially less variation than among the diesels. Thus derating caused an increase in $NO_x$ in only one engine. In general, 25 percent derating reduces $NO_x$ emissions 20 to 35 percent and increases fuel consumption 2 to 8 percent. Derating by 50 percent produces $NO_x$ reductions of 30 to 65 percent, but at the same time fuel usage goes up 10 to 30 percent. In general, small amounts (25 percent or less) of derating appear effective in reducing $NO_x$ emissions from dual fuel units, and such reductions are accompanied by fuel penalties of less than 8 percent.

Figure 4-25(c) shows that the derating of gas engines produces a wide range of $NO_x$ reductions. In general, the nonturbocharged engines achieve the largest reductions, since derating has a greater effect on their air-to-fuel ratio. For example, blowers on blower-scavenged units operate at constant speed, independent of load; therefore, as the fuel flow is reduced to decrease output, the air-to-fuel ratio increases, causing a $NO_x$ reduction. The turbocharged engines, in contrast, maintain a more nearly constant air-to-fuel ratio, and consequently, experience less of a $NO_x$ reduction.

Derating does not require additional engine equipment, and the only operational adjustment is to the throttle or governor setting in order to restrict the engine power output. In most cases, this adjustment can be made in the field, although one could presumably equip a new engine with a fuel pump or carburetor whose maximum fuel delivery capacity corresponds to a derated condition. When derated, the engine's efficiency is reduced, and hence, the fuel consumption is increased. Moreover, when an engine is derated, a bigger, more expensive unit must be purchased to satisfy a given power requirement.

### 4.4.2 Retarded Ignition Timing

As mentioned in Chapter 3, combustion is initiated by the injection of fuel oil in a diesel or dual-fuel engine, or by a spark in a natural gas unit. The effect of variations in the timing of this injection or spark discharge is the same for both kinds of engines; that is the event can be described as combustion ignition in both cases. Therefore, this control technique is termed retarded ignition timing, or retard.

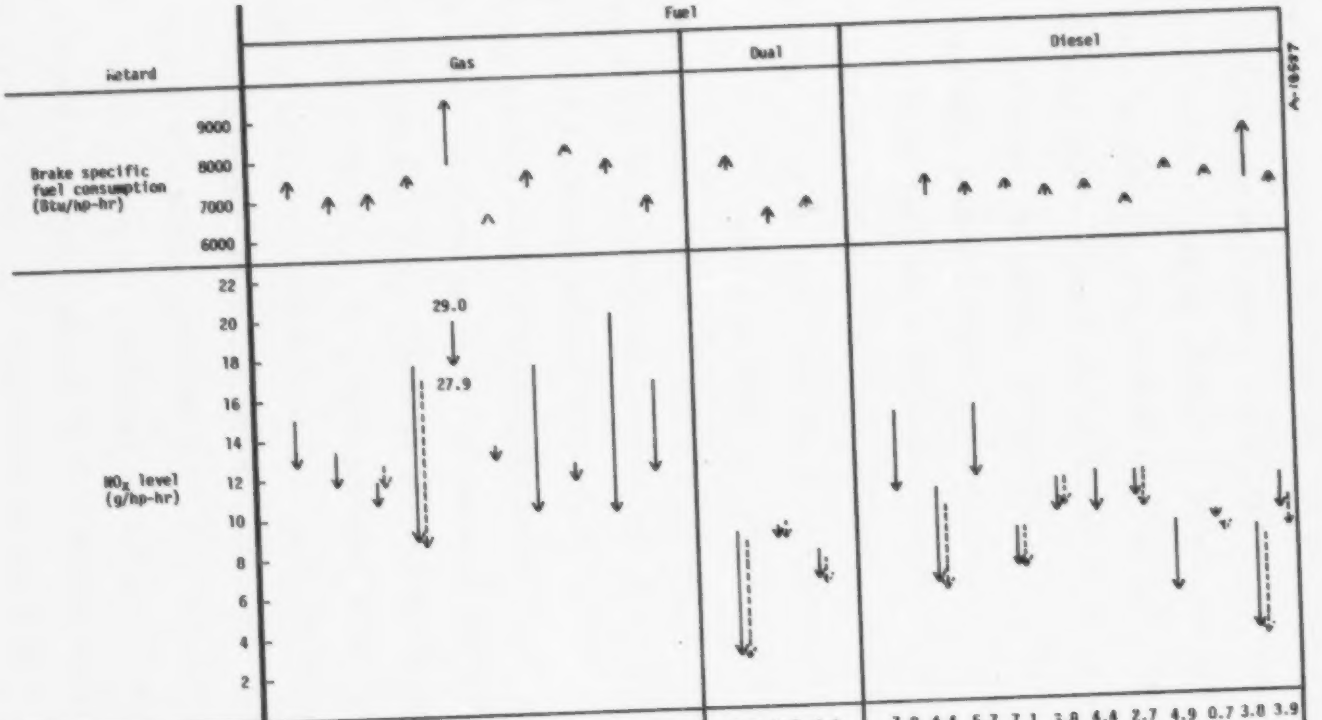
Ignition in a normally adjusted engine is set to occur shortly before the piston reaches its uppermost position (top dead center, or TDC). At TDC the air or air-fuel mixture is compressed to the maximum. The timing of the start of injection or of the spark is given in terms of the number of degrees that the crankshaft must still rotate between this event and the arrival of the piston at TDC. The extent of retard is then expressed in degrees relative to normal ignition. Typical retard values are $2^o$ to $6^o$, depending on the engine. Beyond these levels fuel consumption increases rapidly, power drops, misfiring (erratic ignition) occurs, and smoke from diesel engines becomes excessive(77).

After ignition, the burning combustion gases expand, driving the piston downward. This is called the power stroke. When ignition is retarded, the duration of the combustion process does not change significantly, but rather is initiated closer to TDC and is extended longer into the power stroke. Consequently, the combustion process occurs later during higher exhaust temperatures.

In theory the fuel delivery system in diesel engines could be altered to reduce the duration of injection and thereby decrease the quantity of fuel that is combusted late in the power stroke. Such changes would require increases in the injection pressures above current levels, which are already high. One manufacturer of medium-bore engines, Cummins Engine Co., developed their own high pressure fuel pump for this application because there was no suitable commercial component on the market. Low-volume manufacturers of medium-bore engines, however, depend on outside sources for their fuel pumps and have stated that the inavailability of this critical component restricts their use of retard (e.g., to 2° to 6°) at this time(78). One manufacturer of large-bore engines reported an unsuccessful attempt at increasing injection pressures(79). He found that this higher pressure compressed the puel and expanded the fuel lines and consequently, the fuel injection time was not decreased. Presumably, the test could be conducted with fuel handling equipment so that expansion of the fuel lines, at least, would not prevent a manufacturer from reducing his injection period.

Retarding ignition decreases $NO_x$ formation at the expense of reduced efficiency, thus increasing fuel consumption. Emissions of HC and CO are generally insensitive to retard except in the extreme case where misfiring can occur. That is, the higher exhaust temperatures, which tend to improve the oxidation of any remaining unburned fuel or carbon monoxide, offset the effect of shorter residence times in the cylinder. Smoke in diesel engines, however, increases rapidly after moderate degrees of retard (2° to 6°).

Figure 4-26 presents the level of reduction demonstrated for a range of engine types. Based on these data, the percent of $NO_x$ reduction per degree of retard ranged from 1.2 to 6.9 for naturally aspirated or blower-scavenged engines and from 0.6 to 8.5 for turbocharged engines. Actual reductions due to retardation between 3 to 10 degrees ranged from 0.4 to 7.3 g/hp-hr for all engines. The effect of the control is to consistently reduce the level of $NO_x$ produced, although the magnitude of the reduction can vary considerably between engine types or within an engine category.



| Retard | Gas | | | | | | | | | | Dual | | | Diesel | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Normalized reduction[a] | 2.6 | 3.1 | 2.1 | 16.5 | 0.4 | 0.3 | 4.1 | 1.1 | 5.0 | 3.9 | 8.8 | 1.7 | 6.3 | 7.0 | 4.4 | 5.7 | 7.1 | 3.8 | 4.4 | 2.7 | 4.9 | 0.7 | 3.8 | 3.9 |
| Degrees retard from standard setting | 6 | 4 | 4 | 3 | 9 | 10 | 10 | 6 | 10 | 7 | 8 | 4 | 3 | 4 | 10 | 4 | 3 | 4 | 4 | 4 | 8 | 2 | 16 | 4 |
| Uncontrolled fuel consumption | 7127 | 6775 | 6760 | 7265 | 7840 | 6374 | 7300 | 8139 | 7500 | 6500 | 7040 | 6340 | 6464 | -- | 6679 | 6734 | 6741 | 6677 | 6643 | 6366 | 7147 | 6962 | 6870 | 6643 |
| Percent increase | 5.2 | 4.3 | 4.7 | 3.4 | 15.3 | 1.4 | 4.0 | 0.2 | 3.5 | 5.5 | 10.1 | 1.1 | 2.9 | -- | 7.4 | 1.9 | 2.5 | 0.8 | 2.6 | 1.5 | 6.4 | 1.6 | 18.7 | 3.6 |
| Air charging | BS | TC | TC | TC | NA | TC | TC | TC | TC | TC | TC | TC | TC | BS | TC | TC | TC | TC | TC | TC | TC | TC | TC | TC |
| Strokes/cycle | 2 | 2 | 2 | 2 | 4 | 4 | 4 | 4 | 4 | 4 | 2 | 4 | 4 | 2 | 2 | 2 | 2 | 4 | 4 | 4 | 4 | 4 | 4 | 4 |
| Engine number | 1 | 2 | 64 | 65 | 44 | 4 | 20 | 28 | 45 | 46 | 13 | 6 | 16 | 17 | 12 | 18 | 74 | 5 | 9 | 15 | 26 | 55 | 60 | 8 |

[a] Percent $NO_x$ reduction/degrees retard

Figure 4-26. Effect of retard on $NO_x$ emissions and fuel consumption.

Several manufacturers have investigated the effect of different amounts of retard on $NO_x$ emissions and fuel consumption for diesel, dual fuel, and gas engines. Their results are shown in Figures 4-27 and 4-28. Figure 4-26 shows that the effect of retarding fuel injection on $NO_x$ levels and fuel consumption is similar for different diesel engine types. That is, 4 degrees of retard reduces $NO_x$ from 22 to 30 percent (i.e., 26 ±4 percent), and 8 degrees reduces $NO_x$ 39 to 44 percent. Note that the $NO_x$ reduction per degree of retard decreases for increasing levels of retard. In contrast, fuel penalties increase at a greater rate with increasing retard. Thus, 4 degrees of retard causes a 2-percent fuel penalty, 8 degrees a 6 percent penalty, and 12 degrees a 12-percent penalty. Therefore, maintenance and durability considerations aside, there are diminishing benefits to retarding diesels beyond a certain point, because increases in fuel consumption exceed decreases in $NO_x$ levels.

Figure 4-28 shows similar results for gas and dual-fuel engines, although the data are more scattered. In general, ignition retard for gas engines is not as effective in reducing $NO_x$ levels as it is for diesel and dual fuel engines. For example, 4 degrees of ignition retard gives about a 15 percent $NO_x$ reduction in gas engines as compared to around 25 percent for diesel and dual fuel units. Note that the amount of $NO_x$ reduction remains constant after a certain point for the two naturally aspirated engines, but fuel consumption continues to increase rapidly. In addition, there are practical limits of ignition retard for all gas engines. Spark-ignited engines are more sensitive to ignition timing and, therefore, misfire and exhibit poor transient performance when the ignition timing is not very close to the design point.

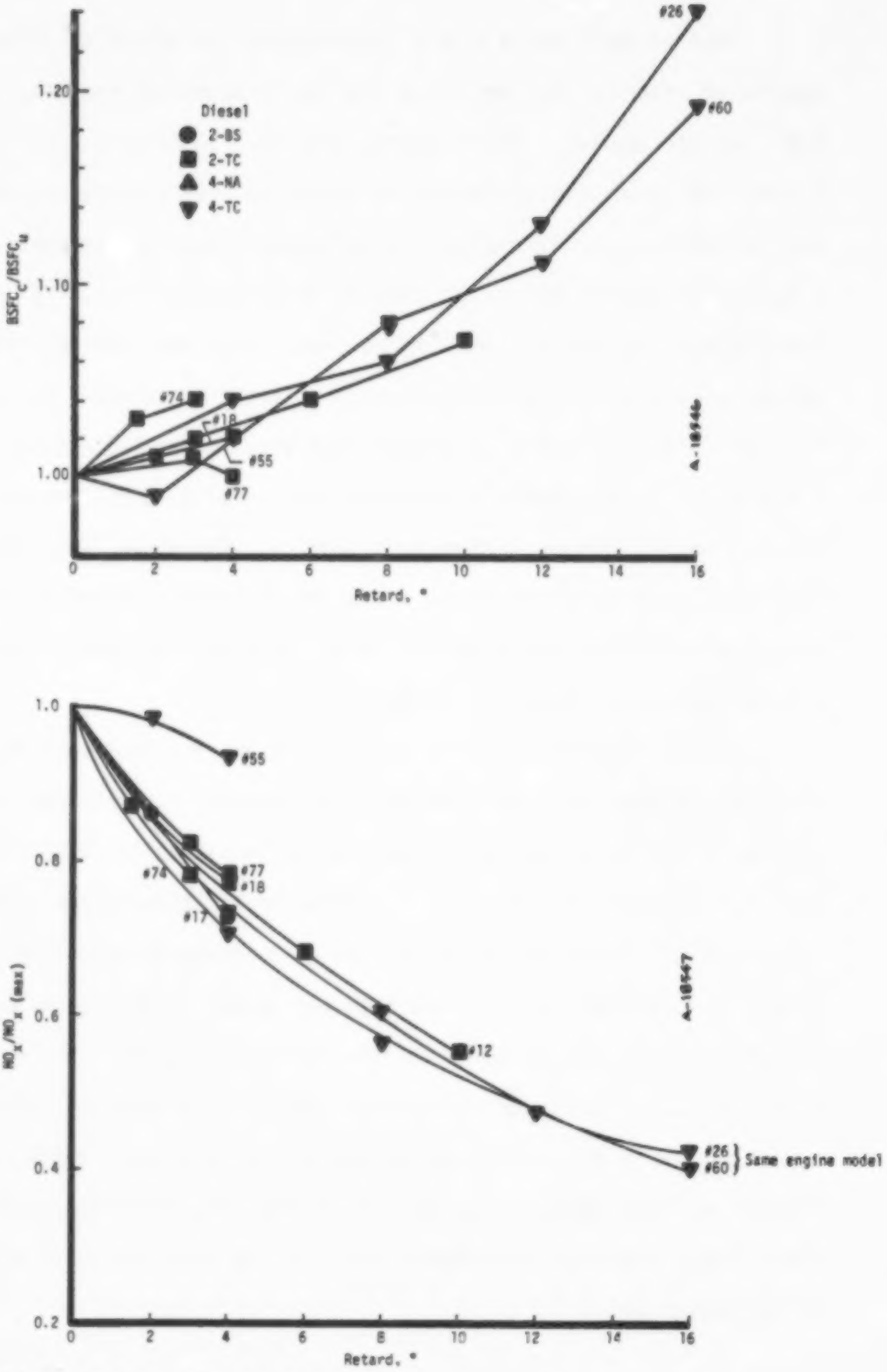


Figure 4-27. Effect of different amounts of retard on $NO_x$ emissions and fuel consumption for diesel engines.

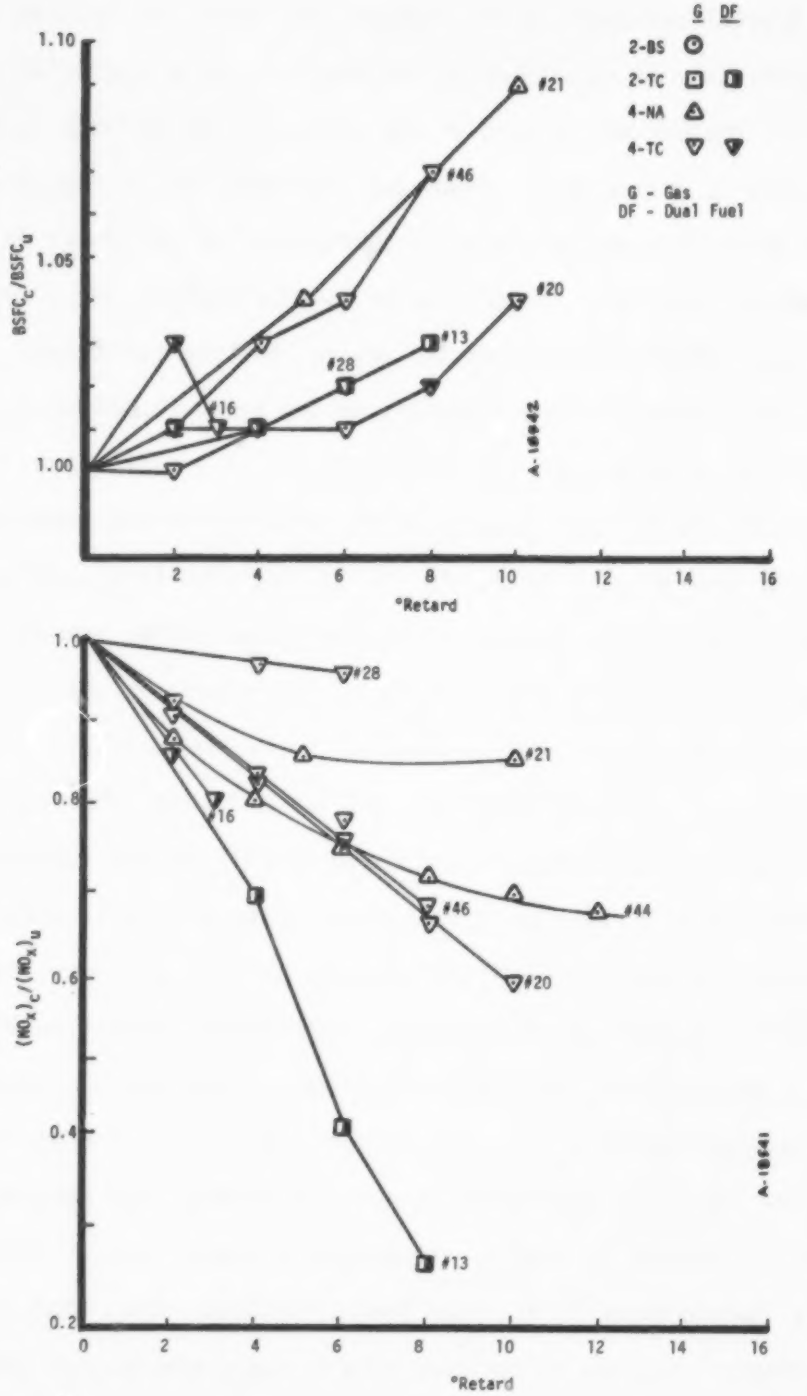


Figure 4-28. Effect of different amounts of retard on $NO_x$ emissions and fuel consumption for gas and dual fuel engines.

Special equipment is not required for injection ignition retard as it involves only an adjustment of the engine spark or injection pump timing. Typically, the nominal setting of the ignition time is fixed by means of hardware items, such as crankshafts. Means are then provided for adjustments, or fine tuning, about this nominal value to compensate for variations in altitude, fuel, engine wear, etc. Manufacturers usually perform this fine tuning service during the production run-in of the engine, but the adjustments also can be made by the operator. This typically occurs every 10,000 hours in the course of normal maintenance, but the setting is actually verified or corrected weekly.

As stated earlier, peak cylinder temperatures and pressures are lowered by retard, and, hence, the thermal and structural loadings are lowered. However, the delayed combustion causes higher exhaust temperatures, which may lead to rapid deterioration of the exhaust valves if the exhaust temperatures exceed the design limits of the valve material. According to a manufacturer of gas-fueled engines, the values in their current production engines can withstand temperatures up to 1300°F, and the turbochargers are limited to 1200°F (these two temperature limits are not inconsistent because the exhaust gas cools between the cylinder exhaust and the turbocharger inlet)(80). Current cylinder exhaust temperatures range from 900°F to 1250°F. Nevertheless, one manufacturer determined that 4° retard of ignition in a dual-fuel engine caused a 25-percent reduction in the maintenance life of his current valve material(81). Another manufacturer reported that his naturally aspirated SI engines are presently operating near their exhaust material limits (1300°F) at rated load conditions. Data from one engine showed that 10 degrees of ignition retard caused the exhaust temperature to increase from 1263° to 1370°F. ($NO_x$ emissions were reduced 17 percent.) Therefore, the application of retard to meet standards of performance may require more frequent engine maintenance or greater initial cost for higher temperature exhaust material.

### 4.4.3 Air-to-Fuel Ratio Changes

The air-to-fuel ratio is defined as the mass flowrate of air ingested by the engine divided by the mass flowrate of fuel consumed. This ratio is termed stoichiometric if precisely enough oxygen is present in the mixture to completely oxidize the fuel. When the ratio is greater than stoichiometric, excess air (oxygen) is present, and the mixture is referred to as lean. Conversely, a lower than stoichiometric ratio is commonly called fuel rich, or simply rich, because more fuel is present in the mixture than can be completely burned.

The maximum $NO_x$ and minimum HC and CO emissions will generally occur at an air-to-fuel ratio slightly leaner than stoichiometric. Although maximum flame temperatures occur at less than stoichiometric ratios, maximum $NO_x$ levels do not occur until lean A/F ratios when oxygen availability is increased. Perfect mixing of the air and the fuel never occurs in existing engines; therefore, some excess air is necessary for complete combustion and minimum HC and CO emissions. These relationships are shown in Figure 4-29(82) for a gasoline-fueled automobile engine. Similar curves apply to diesel- and gas-fired units, with different peak levels for the various curves and shifts in the air-to-fuel ratio that correspond to peak $NO_x$ generation. When the engine is operated rich, HC emissions rise sharply because the available oxygen is no longer sufficient for complete combustion of the fuel. The lack of oxygen for combustion also means it is not present for $NO_x$ formation and so, despite the high cylinder temperatures, $NO_x$ formation will drop sharply at increasingly rich mixtures.

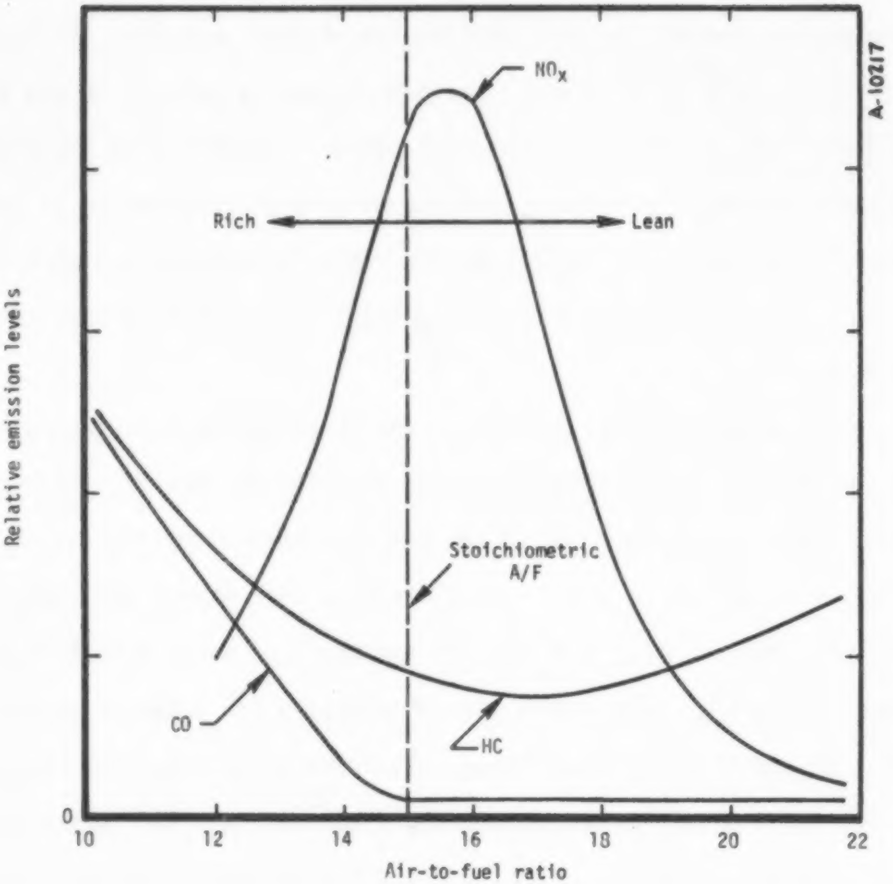


Figure 4-29. Effect of A/F ratio on emissions of a gasoline engine (Reference 82).

If the ratio is varied in the lean direction, the oxygen availability will increase but so will the capability of the air and products of combustion mixture to absorb heat. Consequently, the peak temperature will fall, resulting in lower $NO_x$ formation rates. HC emissions will rise with leaner mixtures due to the lower temperatures which result in increased quenching along the cylinder walls. However, since this effect will be counterbalanced, in part, by the leaner mixtures which occupy the quench layer, the HC rise rate will not be as sharp as it is with increasingly rich mixtures.

CO emissions, which are primarily a function of oxygen availability and only secondarily of temperature, show a pronounced rise as the mixture becomes richer than stoichiometric, but little variation as it becomes leaner.

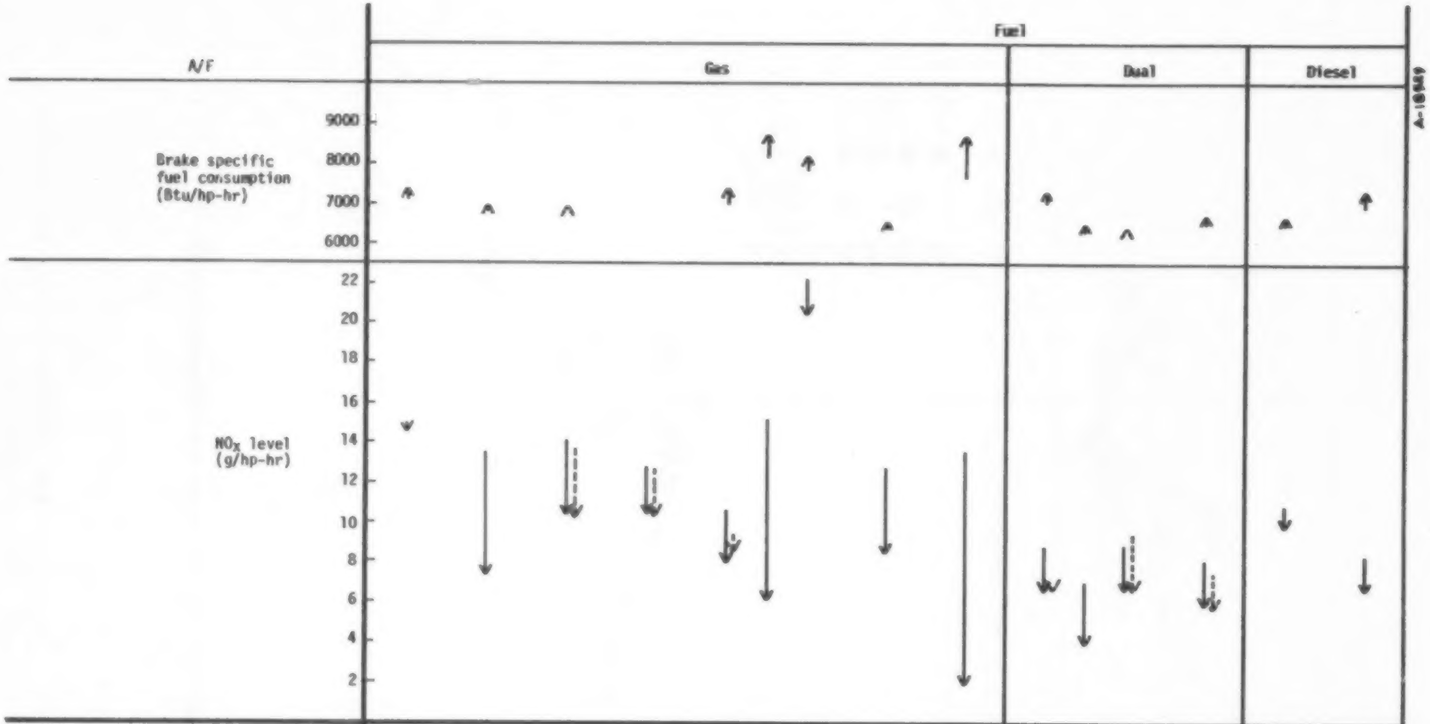
To understand the potential effect of air-to-fuel ratio adjustments on emissions from a particular engine type, one must first examine the ratios at which the engines normally operate. The most important engine characteristic that determines these ratios is the air and fuel charging system.

In injection type engines, which include all diesel and many dual fuel and gas varieties, the air-to-fuel ratio for each cylinder can be adjusted by controlling the quantity of fuel that enters each cylinder. Therefore, these engines can be operated in the lean region where combustion is most efficient and fuel consumption is optimum. On the other hand, carbureted engines are beset by large variations in cylinder to cylinder air-to-fuel ratios(83). Of necessity they must operate near the stoichiometric ratio to insure that no individual cylinder receives a charge which is too lean to ignite (i.e., exceeds the lean misfire limit). Furthermore, A/F ratios cannot be increased too far in turbocharged, spark-ignited engines if severe misfiring and detonation are to be avoided.

The most practical use of air-to-fuel ratio adjustment as a control technique is to change the setting toward leaner operation, since increasingly rich mixture operation causes substantial penalties in both HC emissions and fuel consumption. This technique can be implemented best on injection type engines. Carbureted engines will require better control of the air-to-fuel ratio between cylinders before they can operate at leaner A/F ratios. In fact, some current carbureted engines, when adjusted to leaner than normal air-to-fuel ratios (but still rich), tend to increase their $NO_x$ emissions because they are moving towards the peak of the $NO_x$ versus air-to-fuel ratio curve. They are not able to go beyond that point to the lean region without misfiring(84).

The emission levels that can be achieved by air-to-fuel ratio variation are shown in Figure 4-30. Based on these data, the percent reduction in $NO_x$ emissions per percent of increase in air-to-fuel ratio was 0.12 for the one blower-scavenged gas engine, 1.4 to 6.2 for two naturally aspirated units, and 1.4 to 7.1 for the turbocharged gas and dual-fuel units. Actual reductions in gas engines were about 4 to 6 g/hp-hr for all but the blower-scavenged units and 2 to 3 g/hp-hr for the dual fuel units. The two diesel engines displayed smaller decreases in $NO_x$ emissions, 0.7 to 0.9, with the leaner combustion.

Figure 4-31 shows the effect of changes in air-to-fuel ratio (A/F) on $NO_x$ emissions and fuel consumption for diesel, gas, and dual-fuel engines. In general, small changes in A/F (approximately 10 percent) cause a large $NO_x$ reduction (approximately 30 percent) with less than a 5-percent fuel penalty. This is particularly true for natural gas and dual fuel engines which operate



| Fuel | Gas | | | | | | | | | Dual | | | | Diesel | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Normalized reduction[a] | 0.12 | 4.46 | — | — | 5.08 | 6.2 | 1.40 | 3.28 | 7.1 | 1.38 | 3.9 | — | 2.40 | 0.94 | 0.73 |
| A/F ratio, old/new | 62.5/58.2 | 54.7/59.0 | — | — | 32.0/33.4 | 17.6/16.5 | 21.5/17.0 | 32.5/38.7 | 16/14.1 | 31/36 | 27.6/41.8 | — | 34.3/35.6 | 43.7/40.0 | 28.6/21.1 |
| Uncontrolled fuel consumption | 7127 | 6775 | 6780 | 6600 | 6842 | 8180 | 7840 | 6374 | 7590 | 7090 | 6480 | 6340 | 6464 | 6632 | 6870 |
| Percent increase | 1.3 | 1.4 | 0.6 | 2.1 | 5.2 | 5.6 | 4.1 | 2.3 | 13.9 | 2.5 | 2.1 | 0.8 | 2.0 | 1.1 | 9.3 |
| Air charging | BS | TC | TC | TC | TC | NA | NA | TC | TC | TC | TC | TC | TC | TC | TC |
| Strokes/cycle | 2 | 2 | 2 | 2 | 2 | 4 | 4 | 4 | 4 | 2 | 2 | 4 | 4 | 2 | 4 |
| Engine number | 1 | 2 | 66 | 67[b] | 69 | 21 | 44 | 4 | 29 | 13 | 78 | 6 | 16 | 77 | 68 |

[a] Percent NO$_X$ reduction/percent change in A/F.

[b] Derate 7%

Figure 4-30. Effect of A/F on NO$_X$ emissions and fuel consumption.

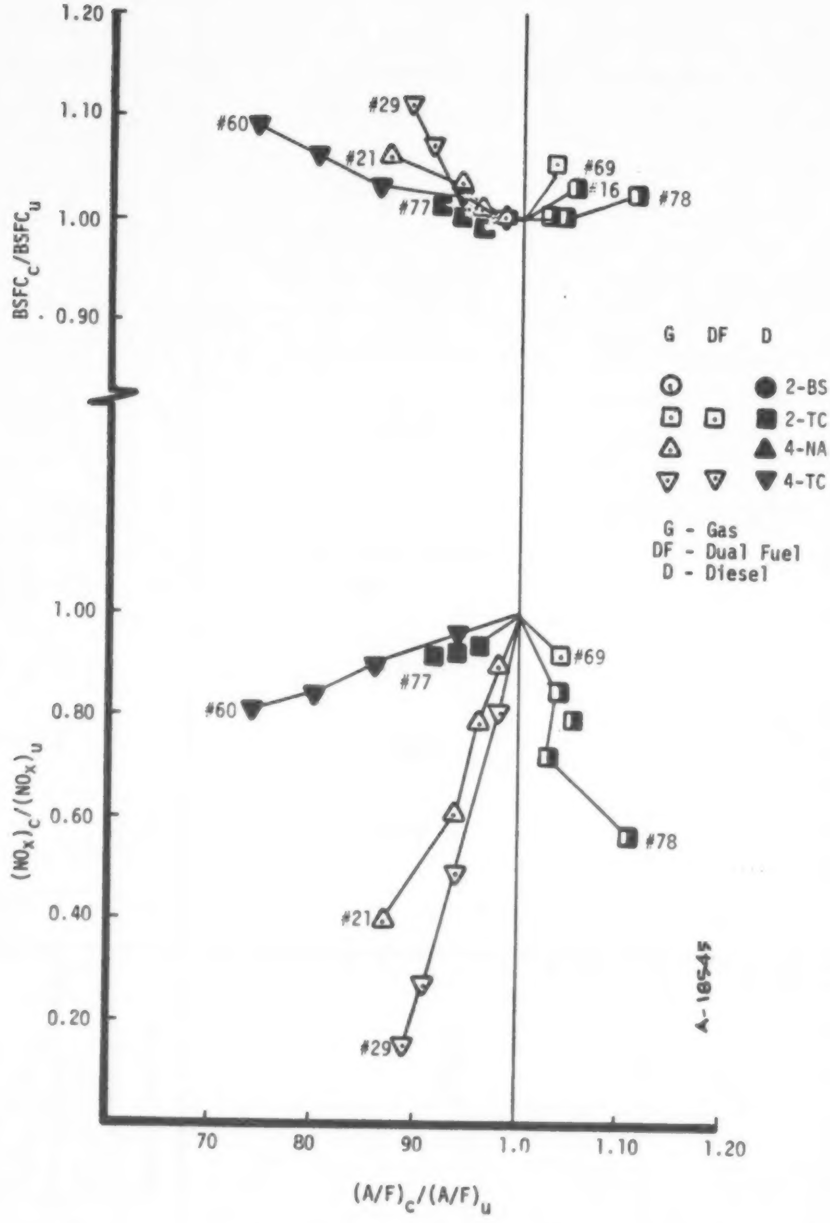


Figure 4-31. Effect of A/F changes on $NO_x$ emissions and fuel consumption.

near stoichiometric (ie., peak $NO_x$ formation). Note that engine numbers 29 and 60 are carbureted and, therefore, are designed to run on the rich side of the peak on the $NO_x$ vs. A/F ratio curve. Hence, they must decrease their A/F ratio (i.e., move to a richer setting) to achieve $NO_x$ reductions. As Figure 4-31 shows, decreased A/F causes rapid increases in fuel consumption. Increasing the A/F, on the other hand, achieves significant $NO_x$ reductions with more moderate increases in fuel consumption. Note that all of the engines that increased A/F were turbocharged units which already operate at lean (greater than stoichiometric) air-to-fuel ratios. Of course, the maximum increase in A/F is limited by the onset of misfiring, as well as marginal transient performance and poor fuel consumption.

In practice, leaner air-to-fuel ratios are obtained by either reducing the fuel input (essentially derating) or by increasing the air input. Increased airflow is accomplished by installing a turbocharger or replacing an existing turbocharger with one that is designed to deliver more air. For blower-scavenged units larger blowers can be installed, or the existing blower can be operated at higher speeds. These changes, however, tend to be offset by increased parasitic horsepower as well as higher discharge air temperatures.

The mass flowrate of air into a cylinder (whose maximum volume is fixed) can also be increased by raising the density of the incoming air through cooling. Intercoolers are frequently placed between the turbocharger exit and the inlet manifold to partially offset the temperature rise of the air which occurs as it passes through the compressor. (Intercoolers are required on large, turbocharged SI engines to prevent preignition.) Both turbocharging and intercooling have other effects on emissions and will be discussed in Sections 4.4.4 and 4.4.5 respectively.

In the case of carbureted engines, increased air flowrates can be obtained by a change in the shape of the venturi and fuel nozzles in the carburetor(85). However, since current carburetor and intake manifold systems do not distribute a uniform fuel-air mixture to all cylinders, significant changes toward lean combustion may require the adoption of fuel injection in place of carburetion(86). In addition, increases in A/F ratio beyond a certain amount reduce the engine's maximum load capability and some derating is required.

### 4.4.4 Turbocharging

The exhaust gas stream from an IC engine contains energy that is normally lost when this stream is discharged into the atmosphere. Some of this energy can be regained by passing the exhaust through a turbine, which is mechanically coupled to a compressor and uses the energy extracted from the exhaust to increase the pressure of the incoming air. This arrangement, called turbocharging, is pictured schematically in Figure 4-32(87). Turbocharging was originally introduced to overcome problems associated with engine operation at high altitudes where the atmosphere is rarefied. Since it increases the pressure of the incoming air, a larger mass of air can be injected into the cylinder of a turbocharged engine than of a blower-scavenged or naturally aspirated one. Hence, more fuel can be injected into the cylinder (the air-to-fuel ratio for optimum combustion is relatively intensitive to pressure), thereby allowing an engine of given size and weight to produce more power. Although turbochargers are normally designed to increase an engine's output to approximately 1.5 times its original power, they can be used to raise the engine's capacity to 2 to 3 times its naturally aspirated value, provided the basic structure is sufficiently strong.

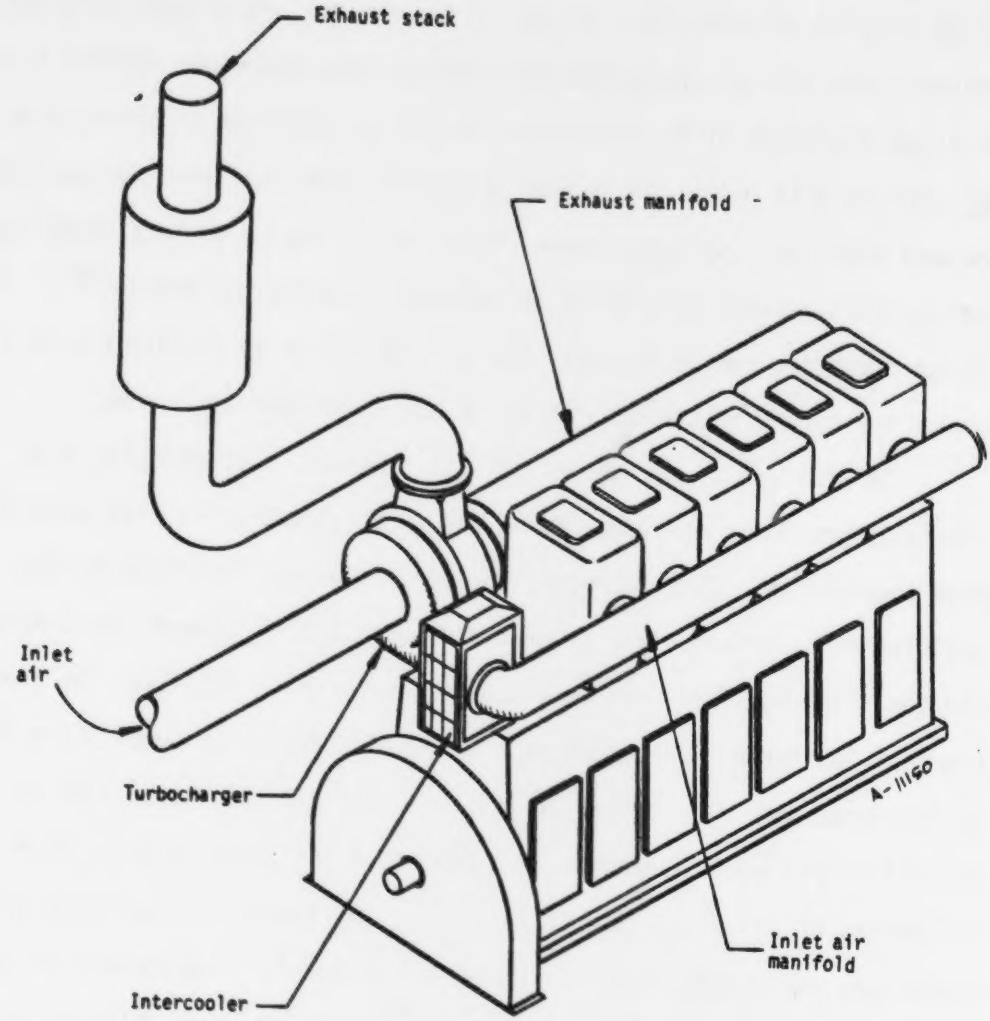
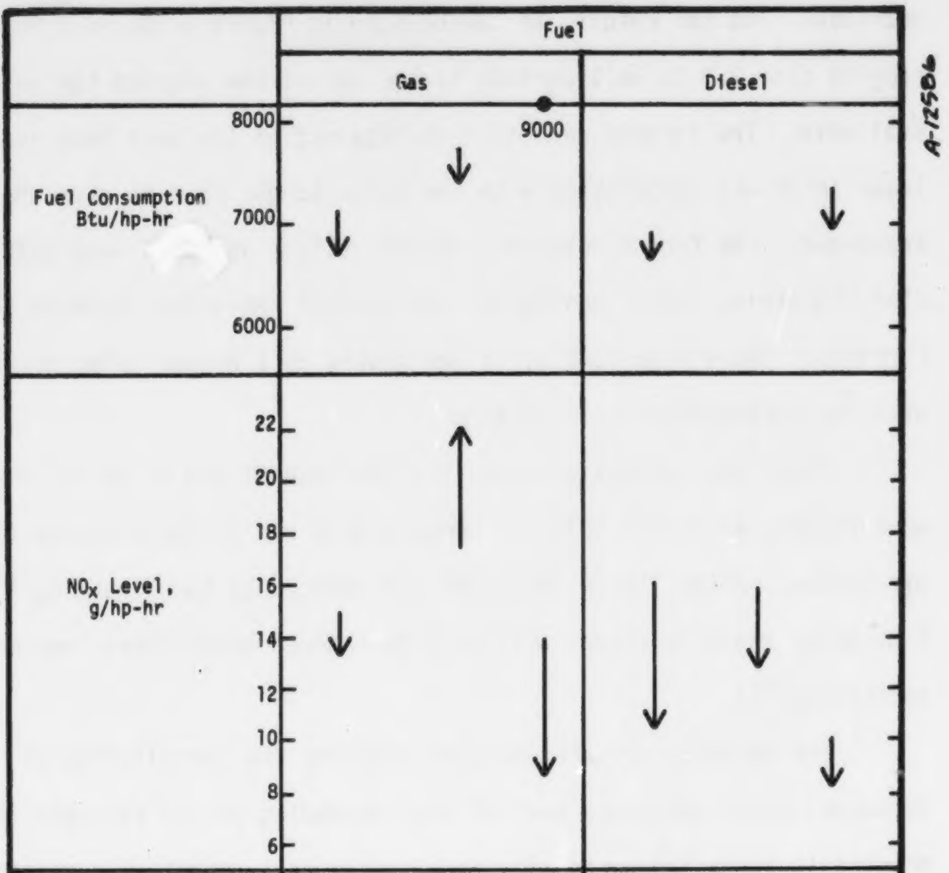


Figure 4-32. Schematic of turbocharged intercooled large-bore engine (Reference 87).

Manufacturers frequently use turbochargers to make engines available that are intermediate in power between existing nonturbocharged units(88). This procedure is more common among manufacturers of medium-bore engines than large-bore units, since most of the larger engines are already turbocharged. Two manufacturers of the medium-bore engines have expressed concern that emission standards which implicitly require the addition of turbochargers to all engines will reduce the number of engine sizes available for purchase because users will no longer have the option of buying engines whose power rating falls between the rating of adjacent turbocharged models(89). Such a change could indirectly increase the cost to a user by forcing him to buy a higher horsepower engine than he would have purchased otherwise.

In addition to increasing the air flowrate, turbocharging also increases the inlet air temperature due to the temperature rise associated with compression. This increase, however, is generally offset by the addition of an intercooler, or heat exchanger placed between the compressor and the inlet manifold (see next subsection for more details). As seen in Figure 4-33, three of the four sets of engines for which comparative data on turbocharged and nonturbocharged are available did not show a rise in the inlet manifold air temperature as a result of the turbocharger. From the standpoint of emissions, and in particular $NO_x$ formation, two combustion parameters may change when a turbocharger is added: temperature and air-to-fuel ratio. If the inlet air temperature rises, the peak cylinder temperature will be correspondingly higher; hence $NO_x$ formation will increase. Conversely, if the temperature is reduced, $NO_x$ formation will be lower. Changes in the air-to-fuel ratio will also change $NO_x$ emissions, depending on the location of the air-to-fuel ratio of the nonturbocharged unit relative to the peak in Figure 4-29 and the direction of the change. In



| | Gas | | | Diesel | | |
|---|---|---|---|---|---|---|
| NO$_x$ Reduction (%)[a] | 10.8 | - 24.9 | 37.1 | 34.5 | 3.3 | 18.0 |
| Air Charging[b] | BS | NA | NA | BS | BS | NA |
| Strokes/Cycle | 2 | 4 | 4 | 2 | 2 | 4 |
| Engine No.[c] | 1/2 | 22/23 | 29 | 11/12 | 17/18 | 35 |
| Manifold Inlet Temperature, °F | 130°/130° | 90°/130° | - | 140°/120° | 100°/100° | - |
| Air-to-Fuel Ratio | 62.5/54.7 | 17.9/25.2 | - | 45.5/37.7 | - | - |

[a] Negative values indicate an emission increase.

[b] Prior to installation of turbocharger.

[c] In each case, engines are the same bore and stroke except for addition of a turbocharger with intercooler.

Figure 4-33. Effect of turbocharging on NO$_x$ emissions.

addition, as the power output is increased, the brake specific emissions will decrease. The net result, as can be seen in Figure 4-33, is a $NO_x$ reduction ranging from 3.3 to 34.5 percent in the two-stroke engines for which data are available. The largest reduction is obtained on the unit that runs at a lower inlet air temperature with the turbocharger than it does when blower scavenged. The four-stroke set, engines 22/23, operates very near to stoichiometric, where leaning the air-to-fuel ratio can increase $NO_x$ formation. More importantly, it was tested at a higher inlet air temperature with the turbocharger than without.

Since the combustion reactions for both CO and HC go to completion more readily at higher cylinder temperatures and in the presence of leaner air-to-fuel ratios, their emissions are reduced by turbocharging(90). Similarly, smoke emissions will also be reduced under these lean mixture conditions(91).

The addition of turbocharging requires the installation of the turbocompressor discussed earlier and, depending on the strength of the originally installed parts, may require the replacement of a number of other engine components (piston rings, connecting rods, wrist pins and cylinder heads) with higher strength parts. This may be necessary because turbocharged engines operate with higher cylinder temperatures and pressures and thus experience higher thermal and structural loads.

Turbocharging is generally offered as an option to most current naturally aspirated or blower-scavenged engines and, as such, is well demonstrated and readily available. As an emission control technique it is most effective if used in conjunction with some means of cooling and compressed air, such as the heat exchanger (intercooler) discussed in the next section.

The addition of the turbocharger (either with or without an intercooler) introduces an operational problem only if the engine must respond to rapidly varying loads and speeds. Stationary engines used in portable air compressors could face such demands, but it is unlikely that the large engines for which standards are now being developed would experience a problem. The problem is one of smoke generation due to rich combustion and arises during acceleration because the fuel flowrate increases much more rapidly than the air flowrate. The air flowrate depends upon the power delivered to the turbocompressor, which depends, in turn, on the power that the turbine can extract from the exhaust gases. The energy carried by these gases increases only when the engine output increases. Hence there is a built-in lag in the system which results in insufficient air for a short time after the fuel flowrate is increased. Some manufacturers have solved this problem by careful control of the fuel rate or by using some of the compressed air from the storage tank (if the engine is used to drive an air compressor)(92). Another attempt at a solution is being tried by one manufacturer of turbochargers, who is currently developing a turbine with two kinds of blades. Some of these extract energy efficiently at low pressures and flowrates while others funtion best at the higher flow conditions corresponding to rated engine operation(93).

Therefore, turbochargers should be considered applicable to all large engines, either because the application does not involve rapid load fluctuations or because of compensatory designs.

### 4.4.5 Reduced Manifold Air Temperature

The installation of a heat exchanger (commonly referred to as an intercooler or aftercooler) between the turbocharger and the intake manifold (see Figure 4-32) normally accompanies the use of turbocharging in engines over 500 hp. The function of the intercooler is to lower the temperature of the intake air after it has been heated by turbocharging (see Section 5.3.5). Decreasing the temperature increases the density, thereby allowing more air to enter the cylinder. This in turn permits higher fuel flowrates and consequently higher power output. Decreasing the inlet temperatures also has the secondary effect of reducing the peak combustion temperature and, hence, the $NO_x$ emissions.

Decreasing manifold air temperature will result in an increase in HC and CO emissions because their respective reduction reactions now proceed at a slower rate. However, some diesel engines display a contradictory trend for CO. Presumably the temperature is still high enough to allow the reaction to proceed to completion, and the increase in power, then, results in a decrease in brake specific emissions.

An intercooler is either provided with, or available as an option on, most current turbocharged engine models. In fact, an intercooler must be used when a carbureted, natural gas engine is turbocharged to prevent the hot air from detonating the air-fuel mixture prior to its entry into the cylinder(94). Additional hardware requirements include the heat exchanger, a circulation device for the cooling medium (either a fan for cooling air, an enlarged water pump for engine coolant water, or a new source of cooling), and a control mechanism to regulate the degree of cooling.

It is also possible to use increased cooling as an emission control in engines that are already turbocharged and intercooled. Depending on the level of cooling necessary and the availability of a suitable low temperature heat sink, this can amount to installation of a larger heat exchanger or the addition of an entire heat exchanger system using air, well or municipal water, or a cooling tower. Self-contained radiators, similar to those found on automobiles, are the most commonly used cooling system for large engines. Large radiators could deliver air to the inlet manifold that is no more than 15°F hotter than the ambient air. Thus, in a hot area where daytime air temperatures frequently reach 100°F, the inlet manifold air temperature could not be reduced below about 115°F. Cooling towers would have to be used in such a location to reach lower inlet air temperatures unless a large supply of cold water were available for once-through cooling. Under typical conditions for a hot and humid location, the turbocharged inlet air temperature of 130°F could be cooled to nearly 100°F with a cooling tower.[6/]

The data in Figure 4-34 are all for an intercooler exit temperature of 100°F. The normalized $NO_x$ reduction (percent $NO_x$ reduction per degree Fahrenheit temperature reduction) was used to interpolate or extrapolate the available data, where necessary, to the 100°F temperature for ease of comparison. This normalized $NO_x$ reduction value ranged from 0.9 to 1.2 for gas engines, 0.6 to 2.2 for dual-fuel units, and from 0.1 to 0.3 for diesel engines. In general, manifold cooling was most effective on gas engines, particularly two-stroke models. This is probably due to the large quantities of scavenging air used by these units. When this air is cooled, it not only reduces peak combustion temperatures directly but also indirectly by cooling the cylinder walls.

Figure 4-35 shows the effect of lowering the manifold air temperature on $NO_x$ emissions and fuel consumption. The results vary and depend on the

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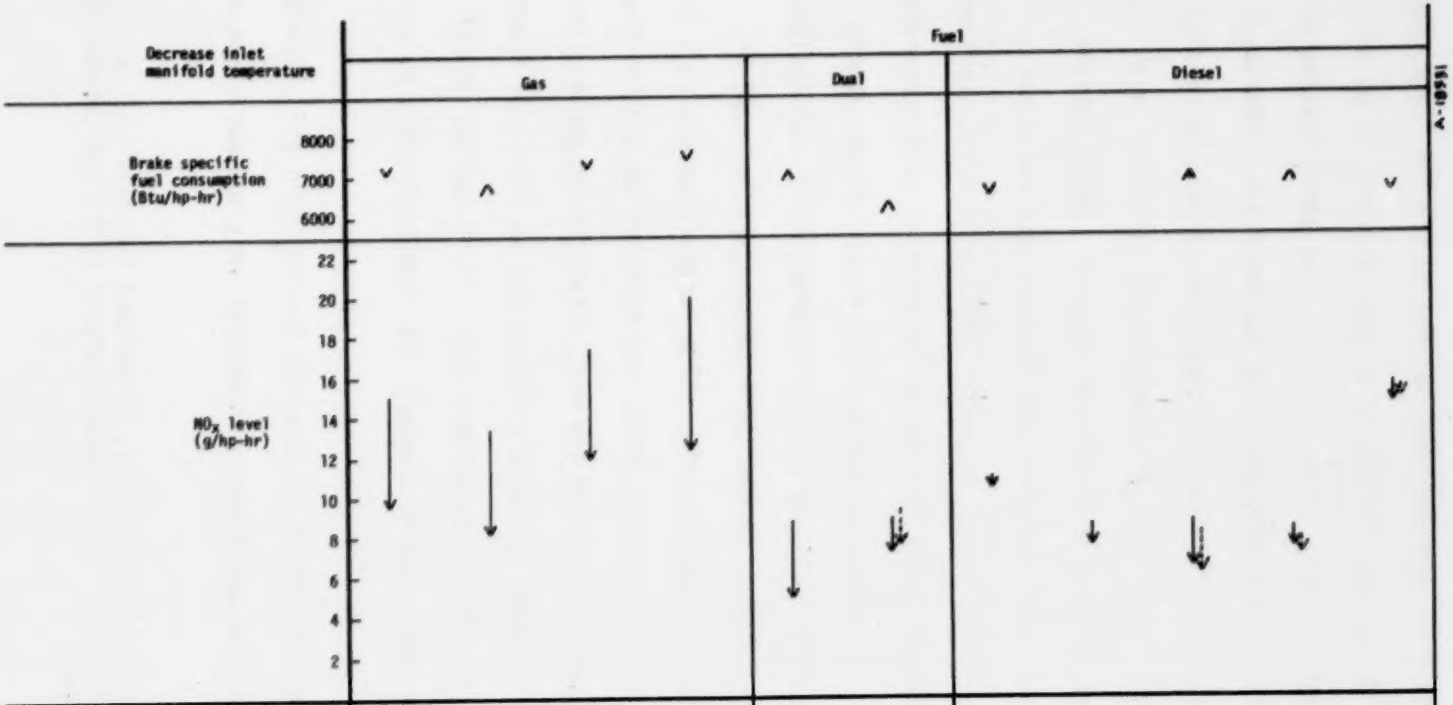
[6/] Based on 100°F ambient air temperature, 90-percent humidity with a resultant minimum intercooler water temperature of 97°F, and typical heat exchanger effectiveness.



| | Gas | | | | Dual | | Diesel | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Normalized reduction[a] | 1.2 | 1.3 | 0.8 | 0.9 | 2.2 | 0.6 | 0.1 | 0.3 | 0.3 | 0.3 | 0.2 |
| Temperature °F | 100 | 100 | 100 | 100 | 100 | 100 | 100 | 105 | 100 | 110 | 100 |
| Temperature reduction | 30 | 30 | 40 | 40 | 20 | 30 | 30 | 35 | 76 | 30 | 30 |
| Uncontrolled fuel consumption | 7127 | 6775 | 7300 | 7500 | 7040 | 6340 | 6677 | 7147 | 6962 | 6870 | 6618 |
| Percent increase | -0.2 | 0.5 | -0.5 | -1.0 | 0.5 | 0.6 | -1.4 | -- | 0.3 | 0.8 | -1.2 |
| Air charging | BS | TC | TC | TC | TC | TC | TC | TC | TC | TC | TC |
| Strokes/cycle | 2 | 2 | 4 | 4 | 2 | 4 | 4 | 4 | 4 | 4 | 4 |
| Engine number | 1 | 2 | 20 | 45 | 13 | 6 | 5 | 26 | 55 | 60 | 68 |

[a] Percent $NO_x$ reduction/temperature reduction.

Figure 4-34. Effect of manifold air temperature reduction on $NO_x$ emissions and fuel consumption.

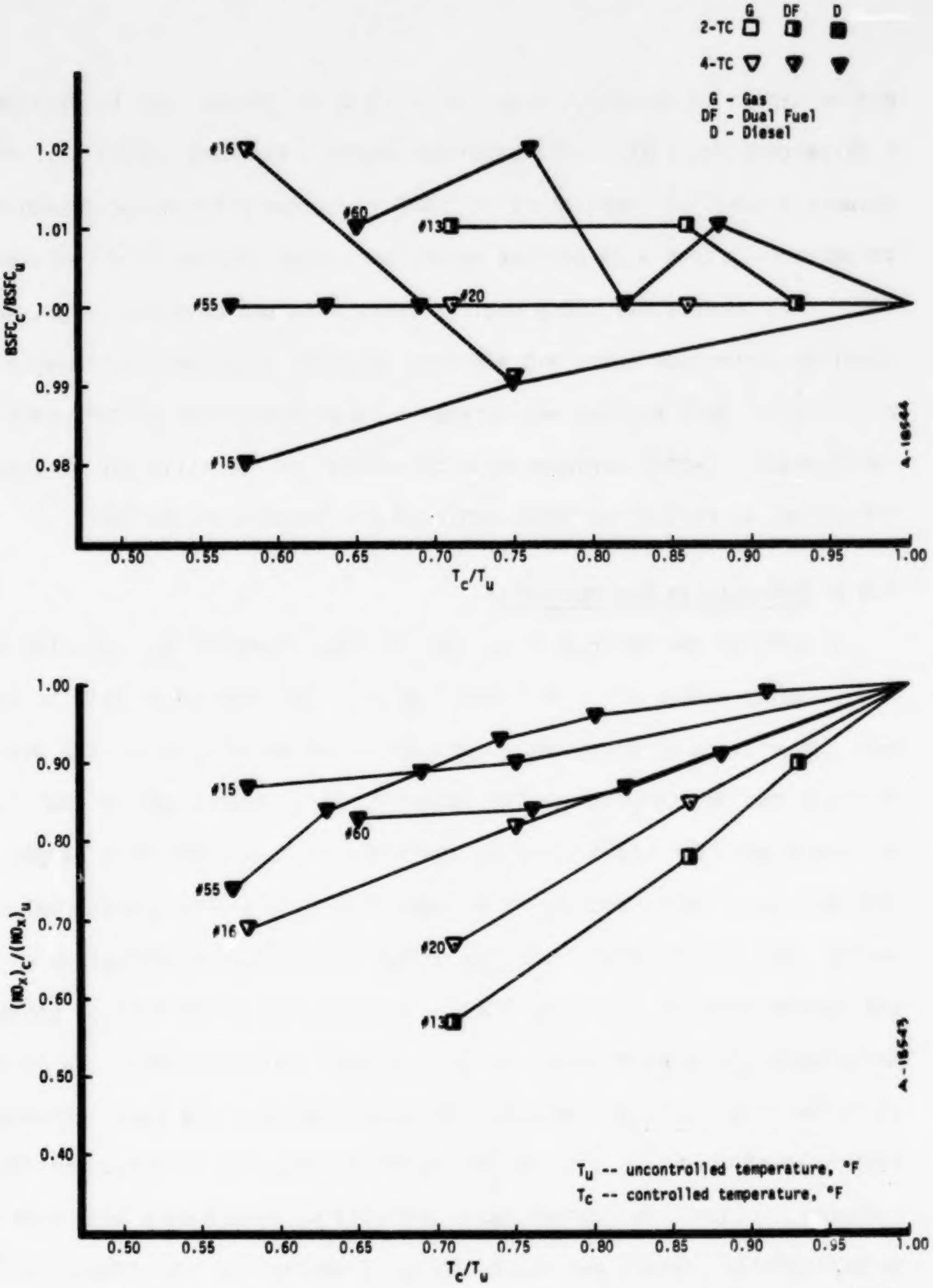


Figure 4-35. Effect of different amounts of manifold air temperature reduction on $NO_x$ emissions and fuel consumption.

engine type. In general, reduction of 10 to 40 percent can be achieved with a 25-percent decrease in the baseline manifold air temperature. If one assumes a baseline temperature of 130°F (a common aftercooler design exhaust temperature), then a 25-percent reduction brings the manifold temperature to 100°F. As noted above, many engines operate in hot climates (e.g., gas pipeline compressor units and electric generating systems in midwestern utilities). Such engines would require large quantities of cold water or refrigeration (which consumes so much energy that derating may be equally effective) to reduce the inlet manifold air temperature further.

### 4.4.6 Exhaust Gas Recirculation

Exhaust gas recirculation (EGR) systems function by replacing some of the incoming excess air with exhaust gases. The purpose of this is to reduce peak temperature by providing a mass which can absorb some of the heat released during combustion. The value of using exhaust gas is that it serves as a heat absorber without making available as much additional oxygen to the nitrogen atoms as turbocharging or operating with a high air-to-fuel ratio would. EGR can be accomplished by either reintroducing exhaust gases into the intake manifold (external EGR) or by restricting the exit of gases that would normally be exhausted from the cylinder (internal EGR). In both cases the effects on pollutant emission are quantitatively the same. Externally recirculated gases can also be cooled before they are reintroduced into the cylinder. Since these cooled gases are able to absorb more heat than unrecirculated exhaust gas would, the peak combustion temperature is lower and less $NO_x$ is generated.

A previous study showed that cooled, external EGR was effective in reducing $NO_x$ from a large-bore, two-stroke blower-scavenged test engine(95).

Although this engine was a two cylinder, older version of a locomotive diesel, trends observed on it should be similar to those that would be observed on modern production engines of this model. The application of 20 percent cooler EGR at rated conditions resulted in a 55 percent $NO_x$ reduction and an increase in smoke to 17 percent opacity.[7/] HC emissions were unchanged and CO emissions were increased 72 percent. By comparison, 20 percent hot EGR resulted in $NO_x$ reductions of 51 percent at rated conditions and smoke increased to 27.5 percent opacity. HC emissions were reduced 17 percent and CO emissions were increased 167 percent.

These same trends have been reported in tests of cooled EGR on truck sized engines which achieved $NO_x$ reductions of 60 percent using 15 percent cooled EGR(96). Smoke increased 300 percent and HC remained relatively independent of EGR rate. CO also increased at EGR rates greater than 30 percent.

Since the oxidation of CO and HC depend upon the availability of excess air and elevated temperatures, one might expect the reduction of both oxygen and temperature by EGR to lead to increased emissions of these two pollutants. However, EGR traps or recirculates some of the unburned, quench layer hydrocarbons in the cylinder. Since these recirculated remnants of the fuel are combusted during the next cylinder firing, HC emissions frequently decrease when EGR is used. Smoke levels increase with EGR due to the reduction in excess air.

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[7/] Twenty percent EGR means that 20 percent of the exhaust gas is returned to the engine intake manifold.

As mentioned earlier, EGR systems can be grouped into two basic categories, internal and external. The application and associated operational problems of each will be discussed in the following sections.

## Internal EGR

With internal EGR, the exhaust products are retained in the cylinder by increasing the exhaust back pressure, retarding the valve timing or by reducing the scavenging airflow in two-stroke engines. The exhaust back pressure can be increased by inserting a restriction in the exhaust gas flow to raise the pressure in the exhaust manifold. This prevents the latter portion of the exhaust charge, which normally exits at the lowest pressure, from leaving the cylinder before the beginning of the next cycle.

Retarded valve timing achieves the same effect by prematurely closing the exhaust valves and opening the intake valves. The early closing of the exhaust valve prevents a complete purge of the combustion gases, while the quickened opening of the inlet valve allows the trapped gases to enter the intake manifold and mix with the air or air-fuel charge before reentering the cylinder. The valve timing in all engine types can be increased by changing the camshaft timing (position of the camshaft, which controls the movement of the valves relative to the crankshaft and, hence, to the piston)(97).

Reducing the scavenging air pressure in two-stroke engines results in an incomplete removal of the exhaust gases by the fresh incoming charge. The scavenging air comes from the air box located below the piston and is driven through the combustion volume by the blower. The scavenging air pressure can be reduced by bleeding off some of the air through a valve located in the side of the air box. This results in incomplete scavenging of the exhaust products which are then trapped in the cylinder and dilute the charge during the next cycle.

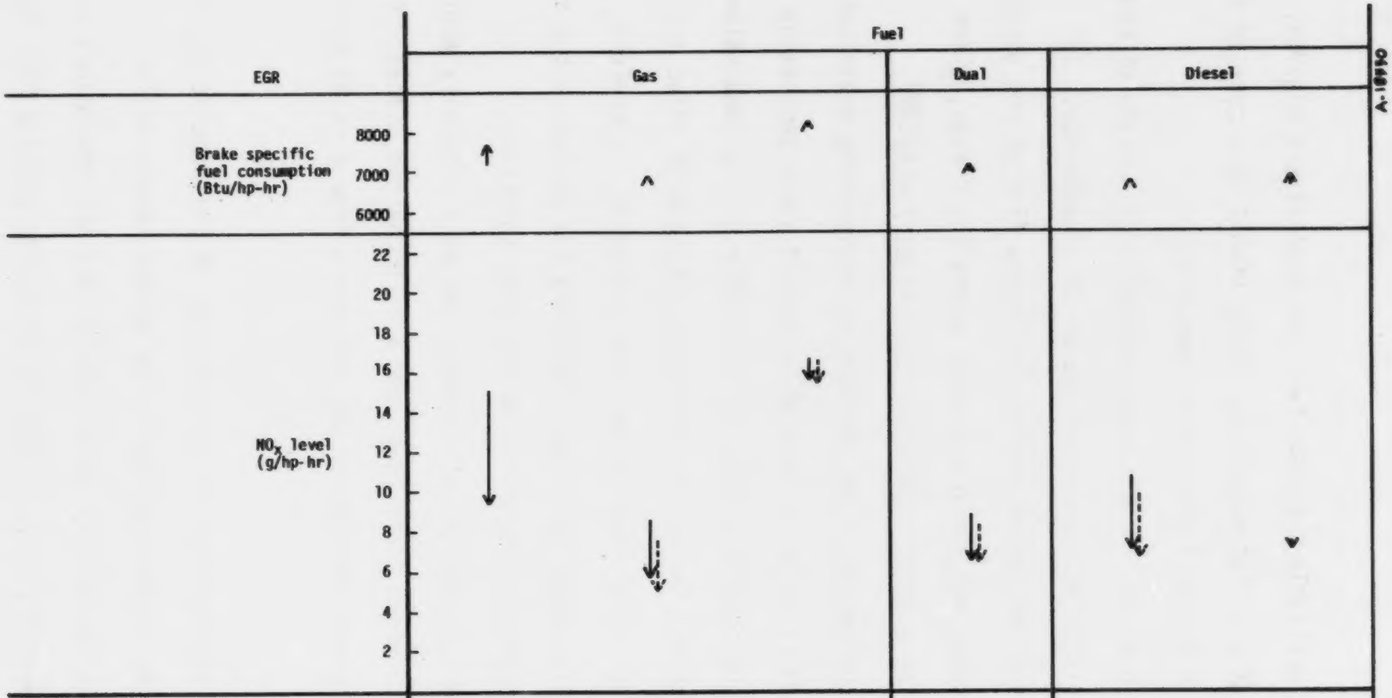
Internal EGR is available for both two- and four-stroke engines, either naturally aspirated or turbocharged. It is usable to a greater extent with turbocharged models due to their leaner operation.

Increased valve overlap has not been adopted for a variety of reasons. First, severe fuel penalties accompany the use of this technique. In addition the inlet valves can become fouled due to the flow of hot dirty exhaust products over them, and the presence of extra inert mass, which contains cold fuel during startup, can make starting difficult(98). Presumably these two problems could be overcome by engineering modifications. A more difficult problem to solve is that of increased smoke generation in diesel engines due to the reduced excess air available during combustion. Unlike engines with external EGR or air box bleed, those with valve overlap cannot reduce the recirculation rate as the load increases. Therefore, they tend to emit smoke at high power settings. Derating the engine to the point where smoke becomes acceptable appears to be the only solution.

Figure 4-36 shows the effects of internal EGR on a naturally aspirated gas engine, a blower-scavenged spark (gas) and a turbocharged diesel. For these three engines internal EGR reduced $NO_x$ emissions from 4 to 37 percent.

## External EGR

External EGR is accomplished by ducting some of the exhaust gas from the exhaust system to the intake manifold. Its effectiveness can be increased by cooling the hot exhaust gases before they are introduced into the inlet manifold. Generally this is done by attaching cooling fins to the recirculating duct to transfer heat from the gases to the environment. The



| EGR | Gas | | | Dual | Diesel | |
|---|---|---|---|---|---|---|
| NO$_x$ reduction (%) | 36.9 | 32.6 | 5.4 | 25.0 | 33.6 | 4.1 |
| External/internal | I | E | I | E | E | I |
| EGR rate (%) | -- | 6.5 | -- | 10.7 | 12.0 | -- |
| Uncontrolled fuel consumption | 7127 | 6845 | 8130 | 7040 | 6679 | 6667 |
| Percent increase | 7.7 | 0.03 | 1.1 | 0.8 | 0.5 | 2.2 |
| Air charging | BS | TC | NA | TC | TC | TC |
| Stroke/cycle | 2 | 2 | 4 | 2 | 2 | 4 |
| Engine number | 1 | 70 | 21 | 13 | 12 | 27 |

Figure 4-36. Effect of exhaust gas recirculation on NO$_x$ emissions and fuel consumption.

data available from tests of external EGR on large engines are included in Figure 4-36, together with the results obtained with internal EGR. The results from three tests on gas, dual fuel, and diesel turbocharged models show reductions varying from 25 to 34 percent. These reductions were obtained with exhaust gas recirculation rates of 6.5 to 12.0 percent. The effect of applying different amounts of EGR on $NO_x$ emissions and fuel consumption is shown in Figure 4-37. $NO_x$ reductions ranged from 10 to 22 percent with 6 percent EGR. In general, fuel consumption remains unchanged for EGR rates less than 12 percent.

Because the available oxygen decreases with increasing load (i.e., increasing fuel flow but nearly constant airflow), it is most efficient to vary the EGR rate with load. Therefore, if the engine is expected to be faced with varying load demands, it should have a control valve in the EGR system regulated by a load sensor. Such a system is pictured schematically in Figure 4-38. Air-to-fuel ratios are typically close to stoichiometric on diesel truck engines when they operate at rated power. Therefore, external EGR systems that have been installed on truck engines to meet California standards for heavy duty vehicles have set their proportional EGR controls to shut the recirculating passage when the engine is run at full throttle. Large-bore engines, on the other hand, run much leaner, even at rated conditions; therefore, they could benefit from EGR though they normally operate at rated power.

The primary durability consideration for external EGR systems, especially when applied to diesel engines, is the accumulation of solid exhaust products in the recirculating system. When EGR is applied to naturally aspirated engines, these deposits build up in the ducts, on any valves used to control the recirculation rate, and possibly on the intake

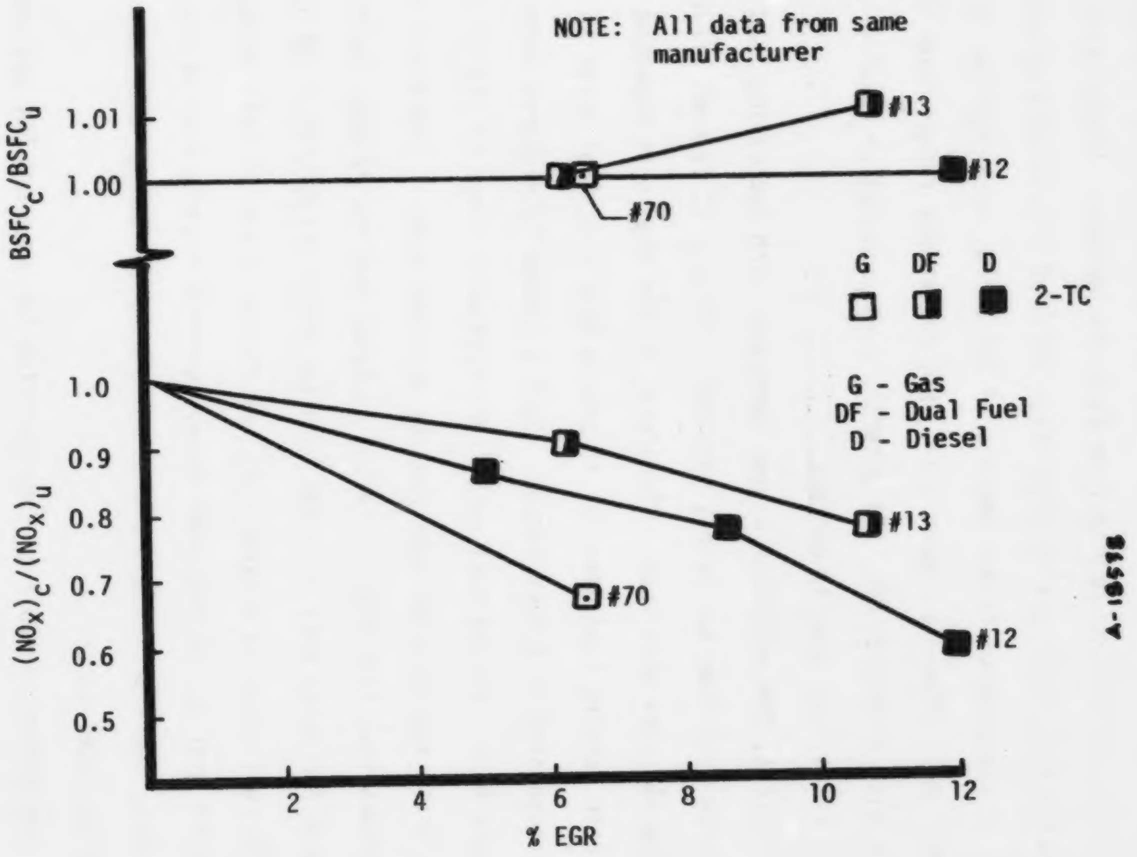


Figure 4-37. Effects of different amounts of EGR on NO$_x$ and fuel consumption for gas, dual fuel, and diesel engines.

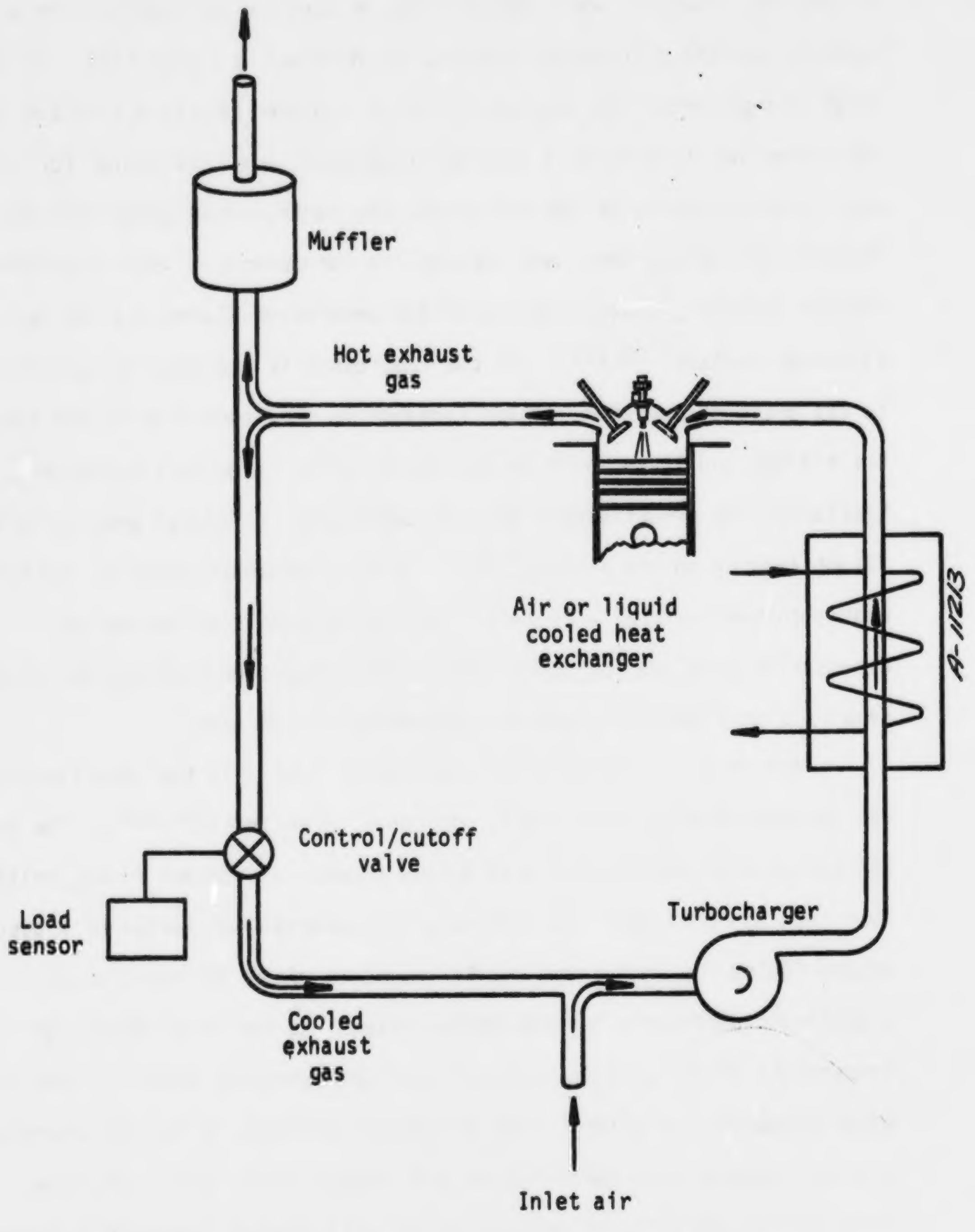


Figure 4-38. Simplified schematic of EGR system for turbocharged intercooled engine.

valves(99). However, when external EGR is used in conjunction with a turbocharged and intercooled engine, the problems are magnified. Since the inlet charge, after the compressor, is at a higher pressure than the exhaust, one either has to provide a separate compressor and intercooler for the recirculating stream or one has to mix the recirculated gases with the incoming air before they pass through the turbocharger. Both approaches have similar problems, namely fouling of the compressor blades and the heat exchanger surface(100,101). If the compressor is designed to operate close to its optimum condition, its performance is very sensitive to the shape of the blades, and these would be changed slightly by deposit build-up. Similarly, the effectiveness of heat exchangers is reduced greatly by any slight coating on the surface(102). In fact, moderate deposits can make the heat exchanger virtually useless. Such deposition problems do not necessarily preclude the use of EGR on these types of engines, but they would require significantly increased maintenance by the user.

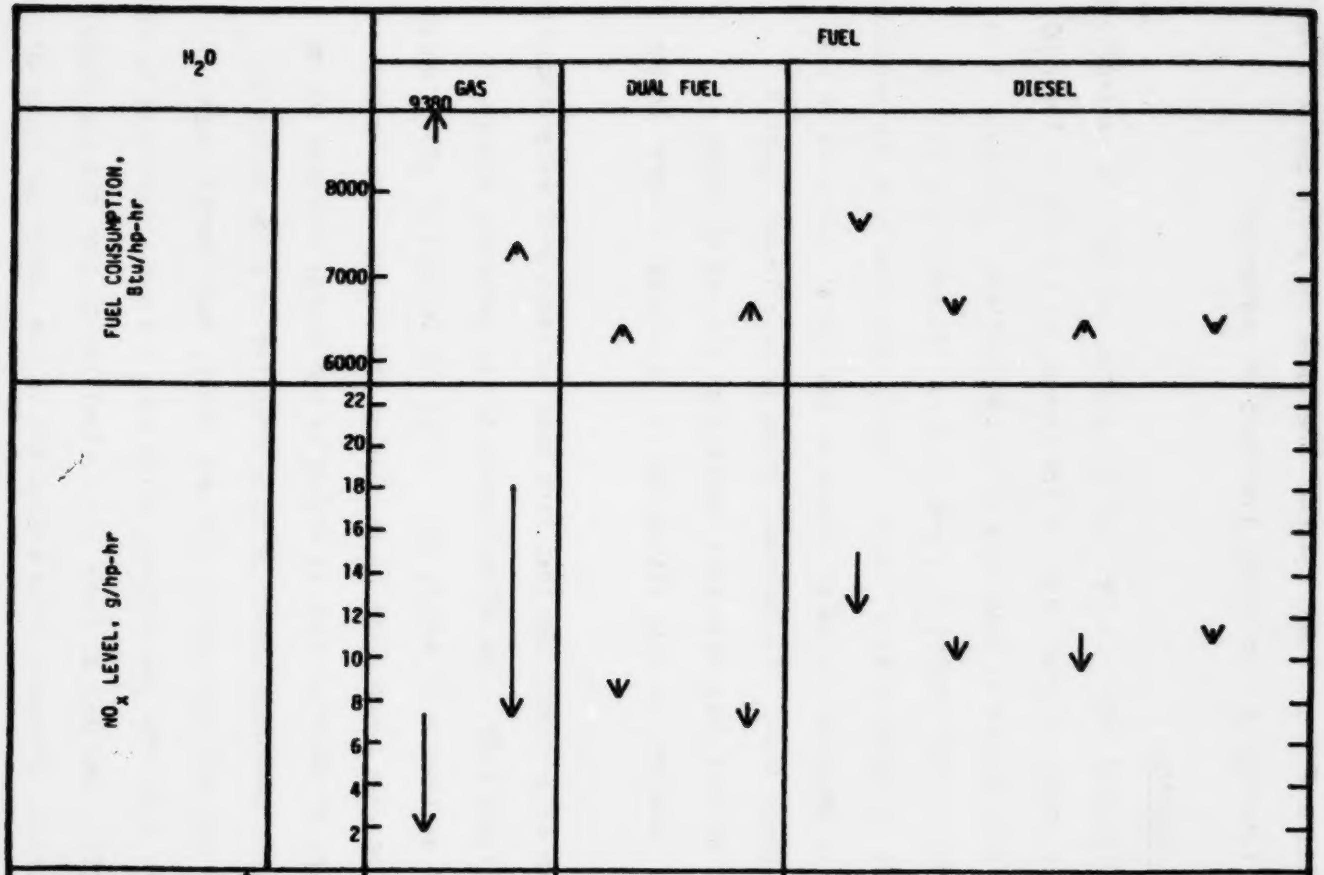
Two studies have been conducted which evaluated the deterioration of EGR systems in heavy duty diesel and gasoline engines(103,104). The external EGR system evaluated in a diesel engine showed no increase in $NO_x$ emissions over 1000 hours of use. In contrast, the external EGR system of a gasoline engine fouled the carburetor and EGR passages after 700 hours of use. The significant difference between these systems was the relatively high temperature of the recycled exhaust gas (approximately 600°F) in the diesel, which prevented the condensation of exhaust products in the air charging system. However some condensation will always occur, and, therefore, trouble-free operation of any EGR system will involve increased maintenance.

One engine manufacturer has also expressed concern over the contamination of lubrication oil in EGR equipped large engines(105). This problem has not occurred in current production automotive EGR systems, presumably because of their lower cylinder pressures and their use of a fuel with a lesser tendency to form sludge (sediment in engine oil).

### 4.4.7 Water Induction

The addition of water to the fuel-air charge has much the same effect as EGR. It increases the inert mass of the charge and results in lower $NO_x$ production through decreased peak combustion temperatures. Although it is easier to insure uniform mixing if steam is used, the water can be more effective if it is injected as a liquid. Energy extracted from the products of combustion to vaporize the water causes an additional reduction in the peak cylinder temperature. Furthermore, steam is usually not readily available, and no test data have been reported on the use of steam. Therefore, the remainder of this discussion is restricted to water in the liquid phase.

Nitrogen oxide reductions that have been achieved with water induction are shown in Figure 4-39. The effectiveness of the technique appears to correlate with an absence of excess air. Since the purpose of adding water is to reduce the peak temperature by increasing the thermal mass in the cylinder (amount of material that is heated by the energy released during combustion), this technique should be more effective in a low excess air system than in one with much excess air and, hence, much thermal mass. As shown in Figure 4-39, the gas engines, which were the most responsive to the addition of water, operated at lower air-to-fuel ratios than did the diesel or dual-fuel units. Presumably the excess air in the latter two types of engines absorbs all the heat that can be transferred to a fluid in the short time between ignition and peak temperature. The combustion reactions for HC



| | GAS | | DUAL FUEL | | DIESEL | | | |
|---|---|---|---|---|---|---|---|---|
| NORMALIZED REDUCTION[a] | 71 | 64 | 68 | 46 | 34 | 42 | 25 | 21 |
| AIR/FUEL[b] | 16[c] | 24.8 | 31.7/31.9 | 34.3/33 | - | 32.6/32.5 | 31.9/31.8 | 31.9/31.7 |
| WATER/FUEL (w/f) | 0.94 | 0.92 | 0.1 | 0.25 | 0.53 | 0.17 | 0.58 | 0.21 |
| AIR CHARGING | NA | TC | TC | TC | BS | TC | TC | TC |
| STROKES/CYCLE | 4 | 4 | 4 | 4 | 2 | 4 | 4 | 4 |
| ENGINE NUMBER | 25 | 20 | 6 | 16 | 17 | 5 | 15 | 15 |

[a] Percent NO$_x$ reduction/water-to-fuel rate.

[b] Where two numbers shown, format is baseline air-to-fuel/controlled air-to-fuel.

[c] Rich air-to-fuel, standard air-to-fuel = 17.5.

Figure 4-39. Effect of water induction on NO$_x$ emissions and fuel consumption.

are inhibited by the lower temperatures, and, therefore, HC emissions increase with water induction(106). Increased quench layer effects at these lower cylinder temperatures may also contribute to the higher HC emission rate. CO emissions are largely unaffected by water induction.

Water induction can be accomplished in a variety of ways ranging from introducing it with the intake air to injecting it directly into the cylinder. These systems vary in both complexity and effectiveness. The effectiveness of the system is strongly a function of the degree of atomization and mixing of the water within the combustion charge. The best systems are those which provide for induction prior to the turbocharger, if present, or spray injection into the inlet manifold or cylinder directly. To control the water flow at various loads, a water pumping and metering system (similar to the fuel system) must be provided.

The reported effectiveness of this control method in reducing $NO_x$ emissions depends almost linearly on the rate at which water is added(107). As shown in Figure 4-40 reasonable reductions (e.g., more than 30 percent) for large-bore engines are obtained only if the water flowrate is at least one-half the fuel flowrate, and significant reductions require that the rate of water addition approximately equal the fuel flowrate(108). Data with water-to-fuel ratios greater than 1.0 are available only for high speed, truck-sized engines, but the same trends should apply to large-bore engines as well. As Figure 4-41(109) illustrates, the effectiveness of water induction (percent $NO_x$ reduction per mass of water) decreases as the water flowrate is increased beyond 1.0. This figure also confirms the earlier statement that the effectiveness of water induction is inversely related to the amount of excess air present; greater $NO_x$ reductions are achieved (at any

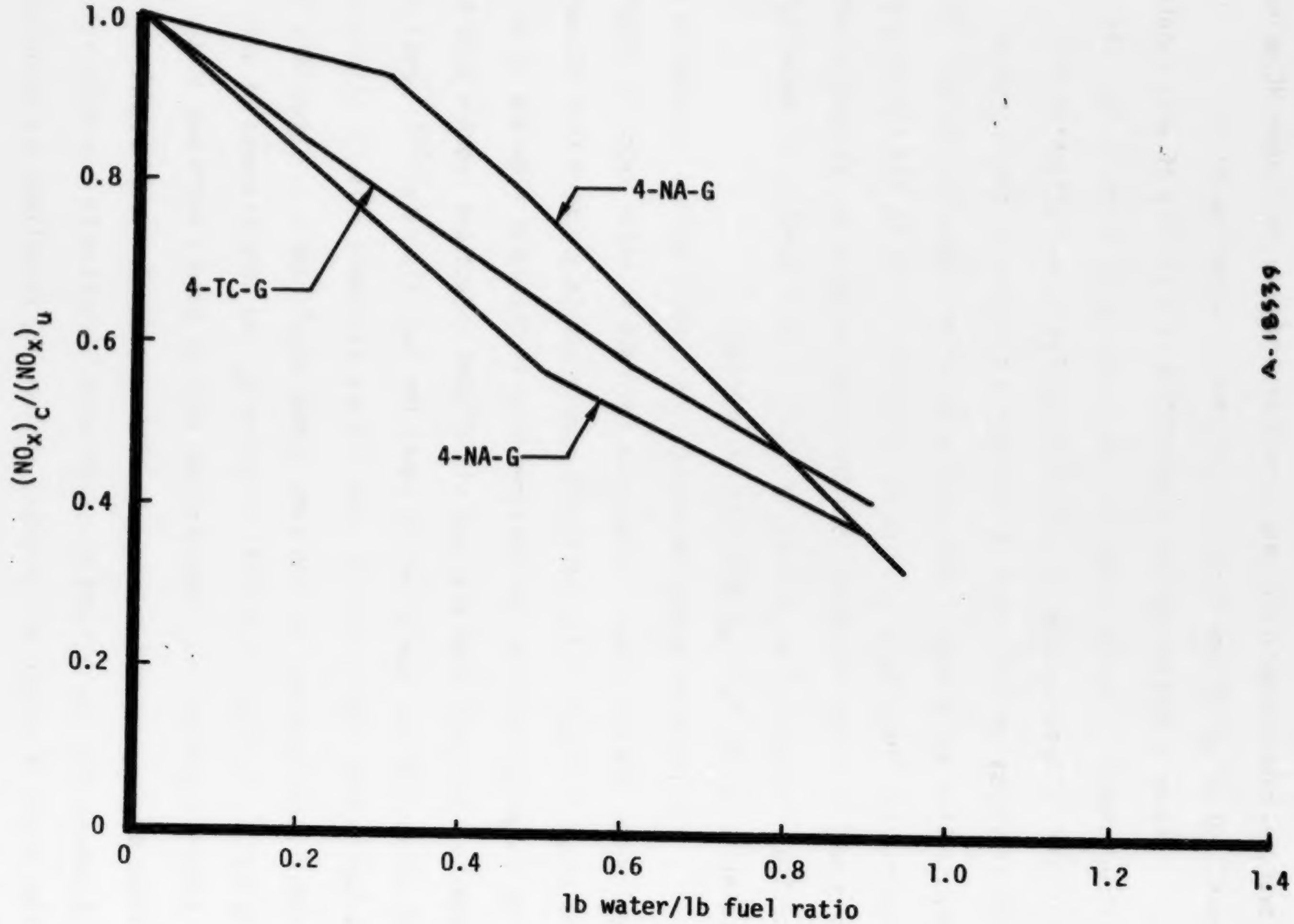


Figure 4-40. Effect of water induction rate on $NO_x$ emissions.

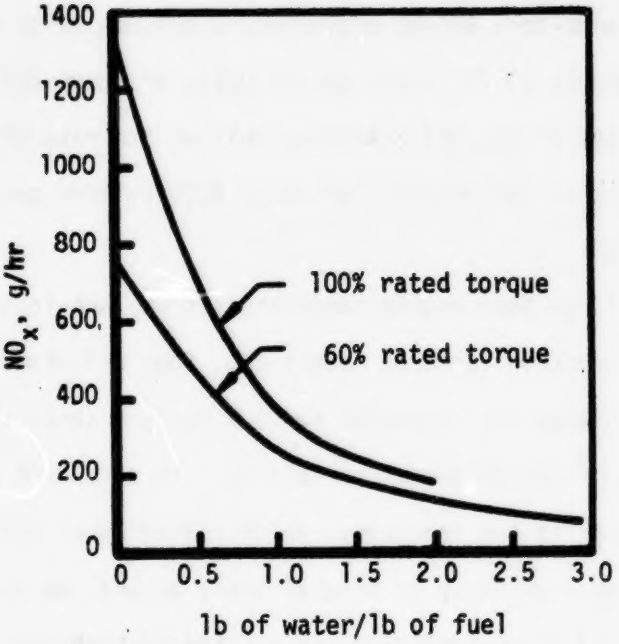


Figure 4-41. Water induction versus $NO_x$ emission for a turbocharged engine at 2200 rpm (Reference 109).

given water-to-fuel ratio) with the engine running at higher load and, consequently, lower air-to-fuel ratio.

Since many engines are used in remote, and frequently arid, locations, water availability may be a problem. Assuming the use of a 1.0 water-to-fuel ratio, a typical large-bore 600-hp engine would consume water at the rate of 0.46 gallons per minute (1.75 liters per minute), or about 660 gallons per day in continuous operation, and a 4300-hp unit at the rate of 2.9 gallons per minute (11.0 liters per minute), or about 4200 gallons per day in continuous operation.

All of the large-bore engine manufacturers who provided results from tests with water induction (General Motors-EMD, Ingersoll-Rand, Cooper-Bessemer, and White Superior) reported serious concern about the adverse effects of this technique on engine durability. This concern was based on observations of water in the crankcase, which contaminated the lubricating oil(110), and on rapid build-up of mineral scale around the valves, water injection nozzles and other components (e.g., the turbocharger) through which the water flowed(111). The one independent researcher who applied this technique to a large-bore engine experienced the same oil contamination(112). Only one long duration test (23 hours) of the use of water induction in a large-bore engine has been conducted to date, and this program used untreated water (400 ppm calcium carbonate)(113). All other tests were short -- usually less than 2 hours of engine running time.

### 4.4.8 Combustion Chamber Geometries

Combustion chamber redesign is the control technique having the greatest potential for reducing $NO_x$ emissions from large-bore engines with little or no loss in efficiency. It is probably also the technique requiring the greatest amount of research and development, and could result in manufacturers changing their engine designs significantly, incurring substantial retooling costs. Several years would be required by manufacturers to perform research and development work and to retool to build engines having redesigned combustion chambers before they could be used to meet a standard of performance for new stationary engines. Some combustion chamber designs which might be applied to reduce $NO_x$ emissions without significant increases in emissions of HC, CO and smoke, or fuel consumption will now be discussed.

The combustion process can be improved by redesigning chamber geometries to increase turbulence, as in high swirl engines(114), by staging as in engines with stratified charge(115), or piston head cavities(116), or by a combination of both, as with "squish lips"(117). High swirl systems use modifications to the design of the cylinder and inlet ports to produce circulatory air motion in the cylinder during combustion. Since these flow patterns are conducive to good air-fuel mixing and, hence, efficient combustion, delayed ignition timing can be used to reduce peak temperatures below those necessary for NO formation with less production of unburned HC and smoke. Unfortunately, no data are available to compare the $NO_x$ emissions of a low- or medium-swirl engine with those from a high-swirl unit when both are retarded as far as possible, i.e., until smoke becomes excessive. The $NO_x$ emissions from the high-swirl engine should be higher than those from the low- or medium-swirl units with corresponding ignition timing and could very well be harder to control; the improved mixing promotes rapid, early combustion and hence high temperatures for a long period of time(118).

All of the staged combustion designs (precombustion chambers, the M-systems, and stratified charge) as shown in Figures 4-42 to 4-44(119-122)

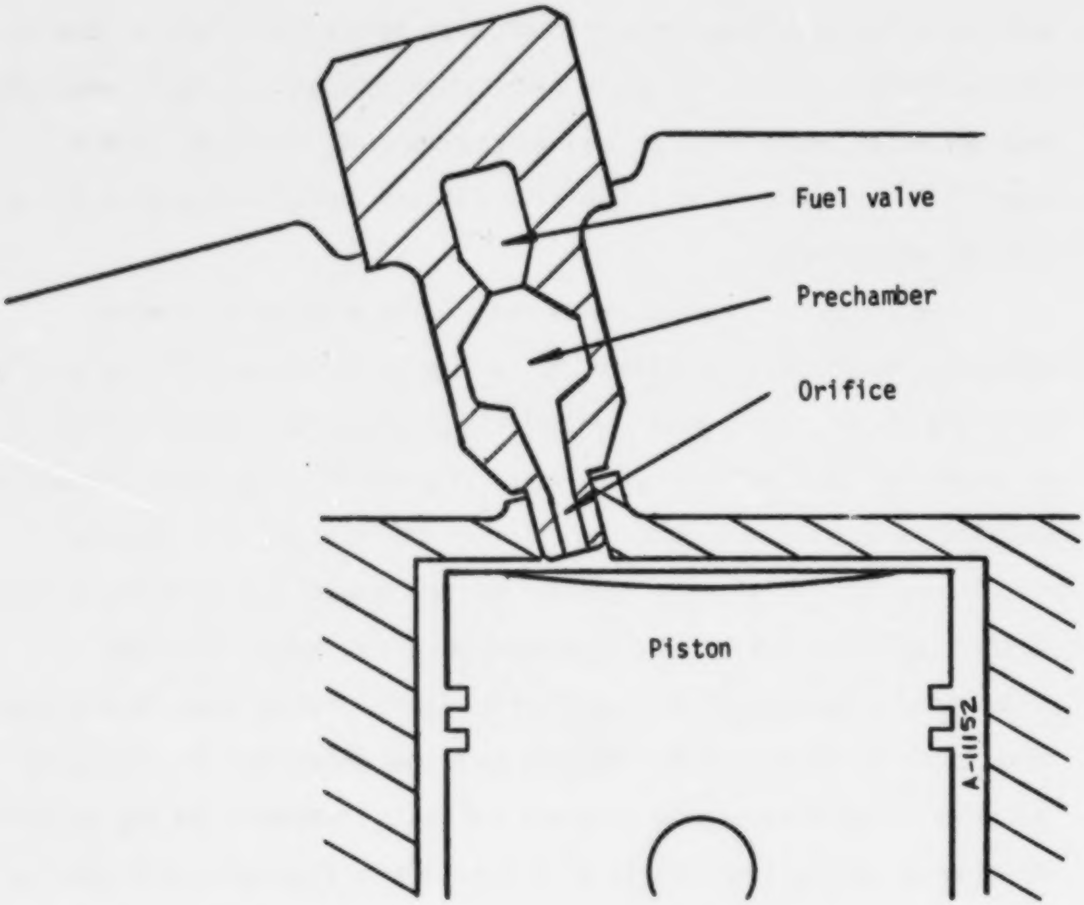


Figure 4-42. Schematic of a current production precombustion chamber for a diesel engine (Reference 119).

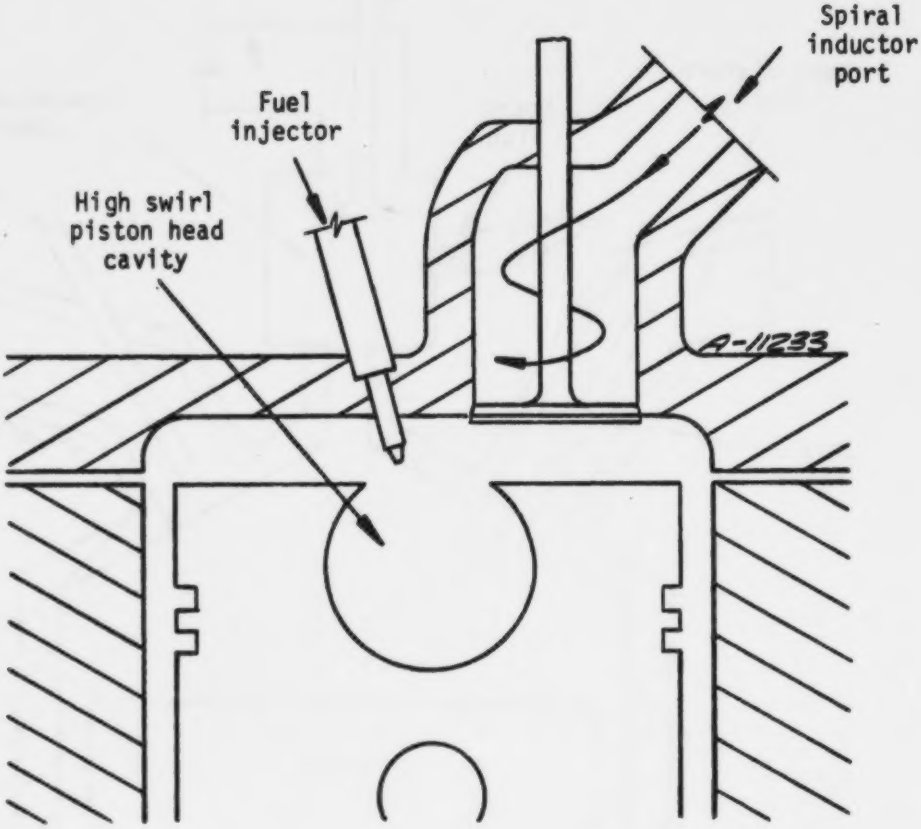


Figure 4-43. Schematic of an "M-system" piston for a diesel engine (Reference 120).

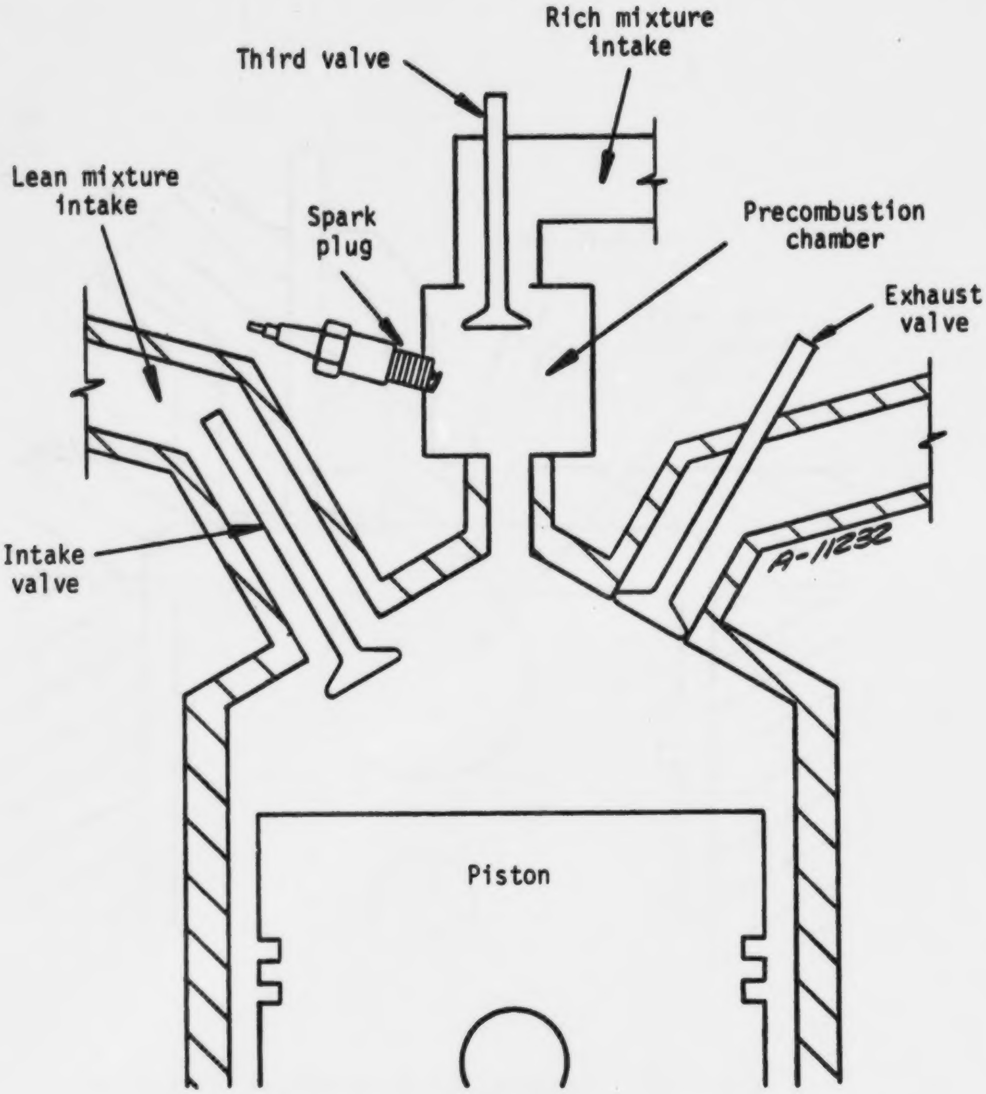


Figure 4-44(a). Schematic of a current production stratified charge gasoline engine (Reference 121).

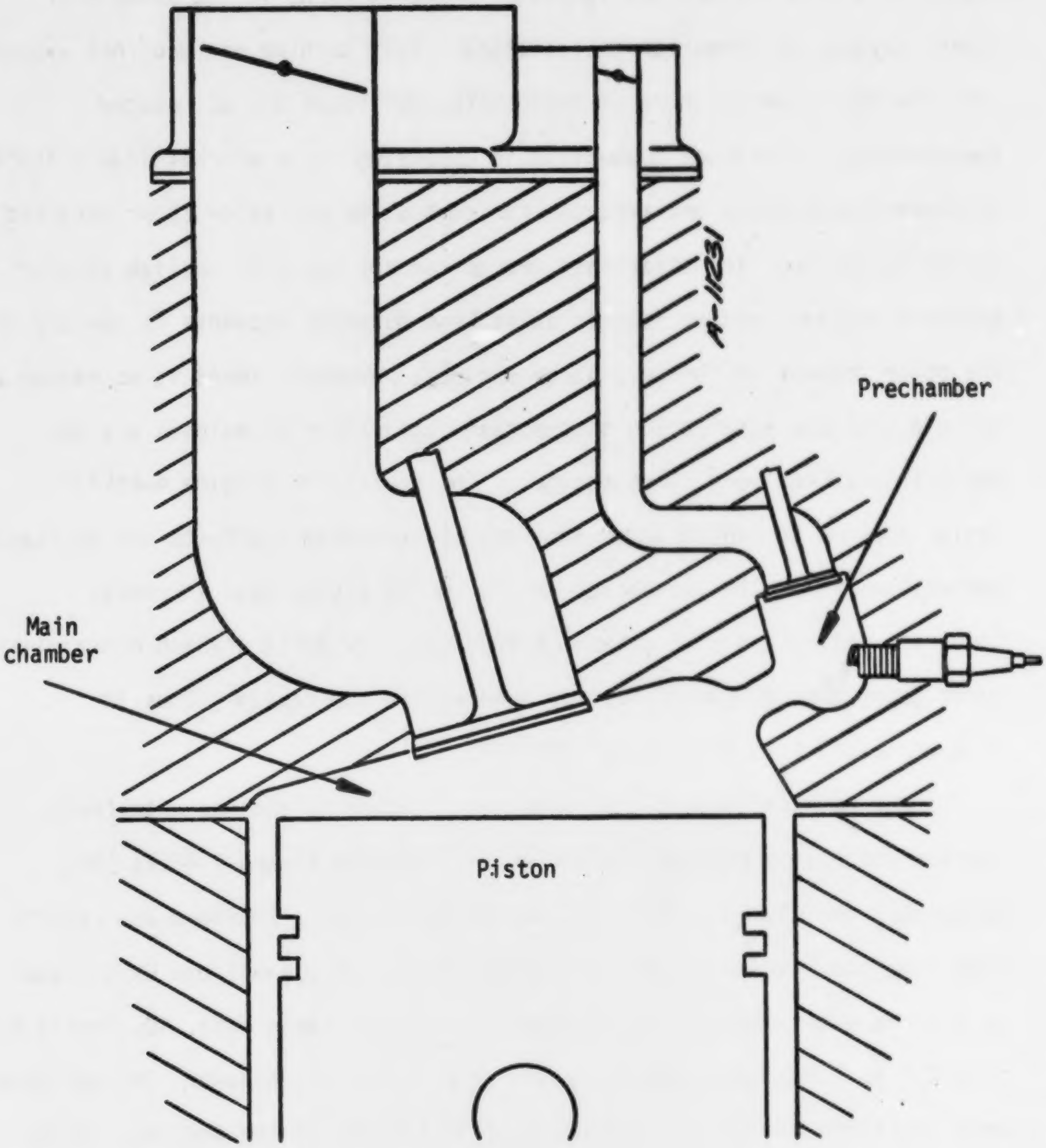


Figure 4-44(b). Schematic of a prototype stratified charge gasoline engine (General Motors Corp. design -- Reference 122).

operate in basically the same manner. The fuel charge is introduced into a cavity as a rich mixture and ignited. Since it burns in the absence of excess oxygen, NO formation is inhibited. This burning mixture then expands into the main chamber where it mixes with additional air at reduced temperatures. Therefore combustion is completed in an overall lean mixture at temperatures which are adequate for combustion but below those required for NO formation. The stratified charge concept has been applied only to gasoline engines, and no reports exist that document attempts to use any of the other designs on the very large engines. However, there is no reason to believe that the application of the basic combustion principles are not applicable to the very large engines. The squish lip (Figure 4-45(123)) design appears to reduce emissions both by aerodynamic effects and by staged combustion(124). The lip-shaped cavity in the piston head increases turbulent mixing and also creates a vertical flow pattern which recirculates burnt gases through the combustion zone within the cavity. Thus it incorporates a form of internal EGR(125).

The emission levels that have been reached by engines with these various combustion chamber geometries are shown in Figure 4-46(a) ($NO_x$ emissions) and Figure 4-46(b) (HC and CO emissions). Included are results from some truck-sized engines and other commercially available units, such as engines with precombustion chambers. Based on these data, $NO_x$ levels ranged from 2.7 to 7.5 g/hp-hr and CO levels from 1.3 to 4.1 g/hp-hr. HC emissions were clustered between 0.2 and 0.6 g/hp-hr for all of the engines. Since these configurations result in approximately 60 percent lower $NO_x$ levels than observed in open chamber engines, and since no large-bore units are currently built with these designs, they can be considered a potential "control technique" for the large engines. Although members of the DEMA have expressed concern

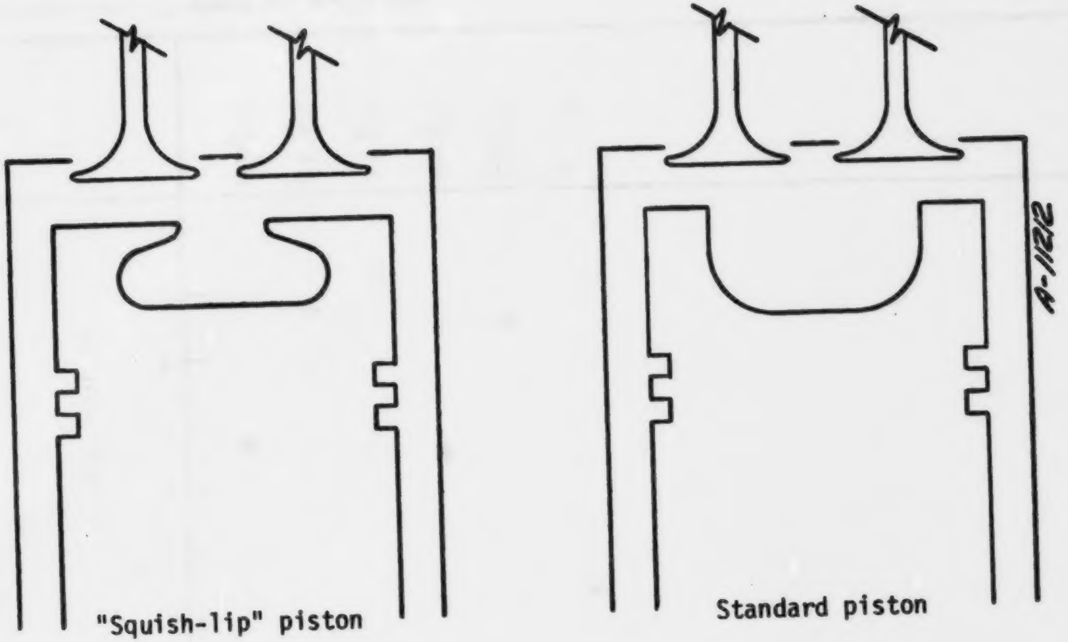
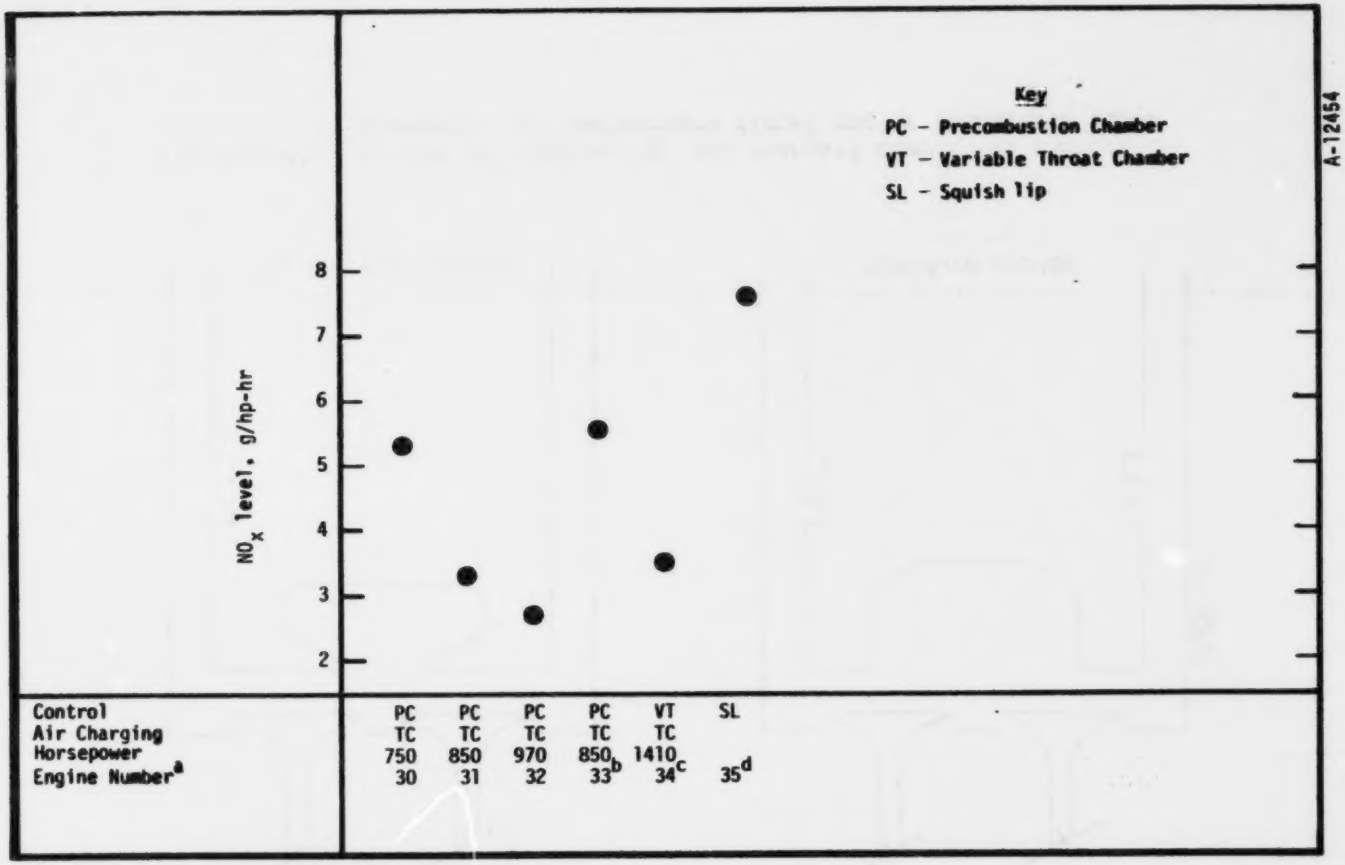


Figure 4-45. Diagram of "squish-lip" and standard combustion chamber geometries for medium-bore diesel engine (Reference 123).



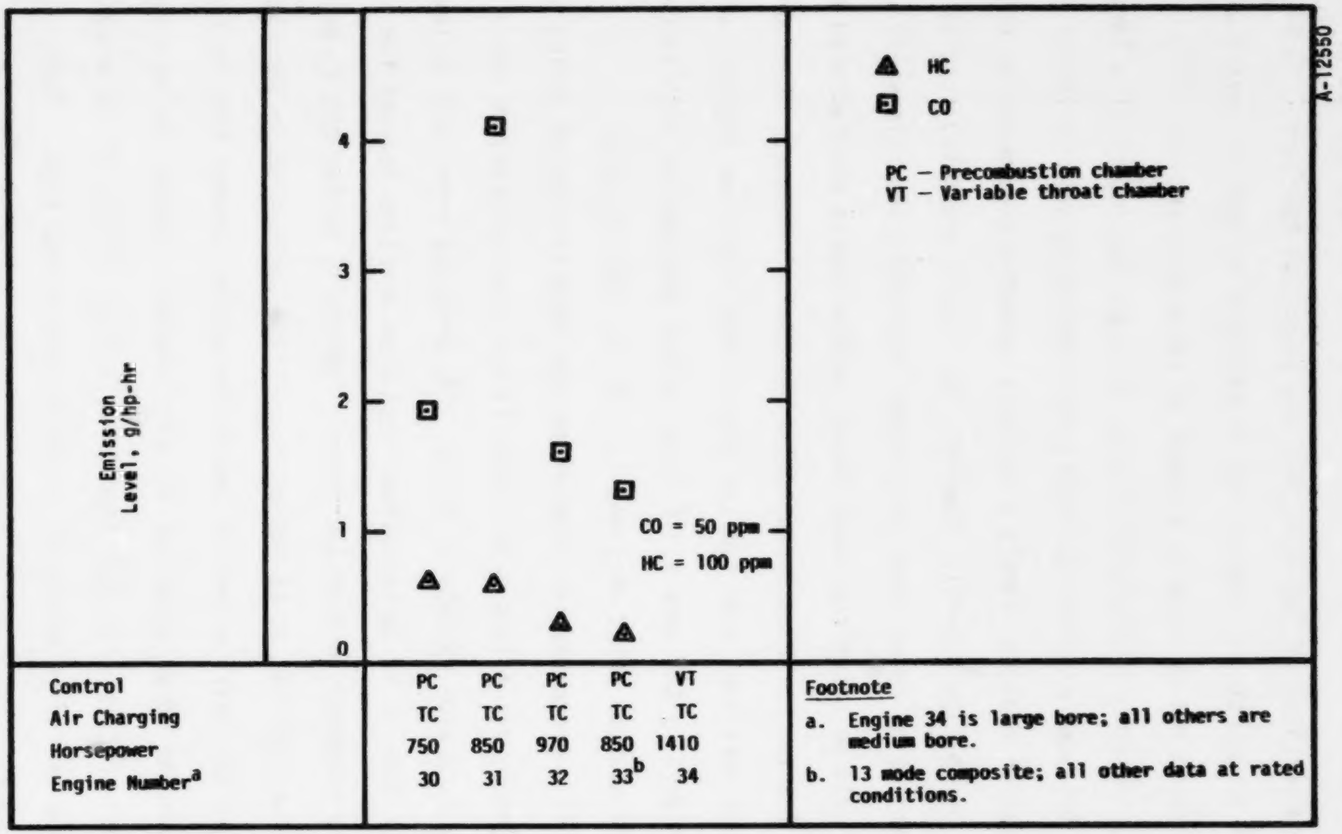
[a] Engine 34 is large bore, all others are medium bore.

[b] 13 mode composite

[c] Fuel consumption = 7200 Btu/hp-hr

[d] 13 model composite, HC + $NO_x$

Figure 4-46(a). $NO_x$ emissions from four-stroke diesel engines with modified combustion chambers.



Footnote

a. Engine 34 is large bore; all others are medium bore.

b. 13 mode composite; all other data at rated conditions.

Figure 4-46(b). HC and CO emissions from four-stroke diesel engines with modified combustion chambers.

about their ability to adapt such designs to their engines(126), the problems appear to be mainly of the kind that can be solved by application of current technology rather than of the kind that are technical barriers. In fact, one manufacturer of squish lip engines has stated that he sees no reason why squish lips could not be used on pistons of large-bore engines(127). Furthermore, the open literature contains at least one report of a large-bore engine that has been modified to incorporate something akin to one of these staged combustion designs, namely a variable geometry precombustion chamber (Engine No. 34 in Figure 4-46). However, the report presented results on a full-scale engine at rated loads only under laboratory conditions(128).

It should be noted that most diesel engine models manufactured by the Caterpillar Tractor Co. have precombustion chambers. However, they consume 5 to 8 percent more fuel than similar size direct injection engines, when both are designed by the same staff (i.e., equal engineering sophistication) to operate at maximum efficiency and to meet the same emission standards(129). Precombustion chamber designs were introduced during a time when fuel economy was a less significant factor than maintenance costs and adaptability to a wide variety of fuels. The pressure rise rate is lower in these engines than it is in the direct injection versions because the combustion is extended in time and space. Therefore, mechanical stresses are also less. Furthermore, it is easier to initiate combustion in the rich mixture within the small volume of the precombustion chamber than in the lean mixture within the large volume of the main chamber. Hence the fuel spray pattern is not so critical, and, therefore, a wide variety of fuels can be burned without having to adhere to a strict maintenance plan. These advantages are now outweighed by the need to conserve fuel. Therefore, Caterpiller is planning to phase out precombustion chamber engines within the next few years, and their competitors plan to meet more stringent standards (e.g., California's 1975 and 1977 heavy duty vehicle standards) without resorting to this design.

### 4.4.9 Catalytic Converters

The automotive industry has expended considerable research and development during the past 5 years for exhaust catalysts to reduce HC and CO emissions. These catalysts, however, are not effective in reducing $NO_x$ emissions. Catalytic converters that reduce $NO_x$ emissions have been developed and applied only recently to 1977 model year cars(130). These systems are called "three-way conversion (TWC) catalysts since they reduce HC, CO, and $NO_x$ emissions simultaneously. These TWC catalysts closely resemble the earlier oxidizing catalyst (precious metals applied to monoliths or pellets) with the addition of rhodium which selectively reduces $NO_x$ to $N_2$. To operate properly, the air-to-fuel ratio of the automobile exhaust under operating conditions must be controlled to a narrow range about stoichio metric. The recent availability of durable ceramic exhaust gas oxygen sensors to determine A/F conditions and control logic systems to regulate the A/F conditions in the exhaust have made possible the implementation of this control system(131). Figure 4-47 illustrates the approximate location of this device on an engine.

These catalytic systems, however, are not applicable to large-bore engines for two reasons: (1) HC and CO emissions from large-bore engines are relatively low in comparison with vehicle level (typically 0.1 to 0.5 g/hp-hr nonmethane HC and 0.2 to 5.0 g/hp-hr CO for a large engine versus 1 to 6 g/hp-hr reactive HC and 30 to 90 g/hp-hr CO for a 100- to 200-hp gasoline fueled engine) (2) A/F conditions in the exhaust of large engines

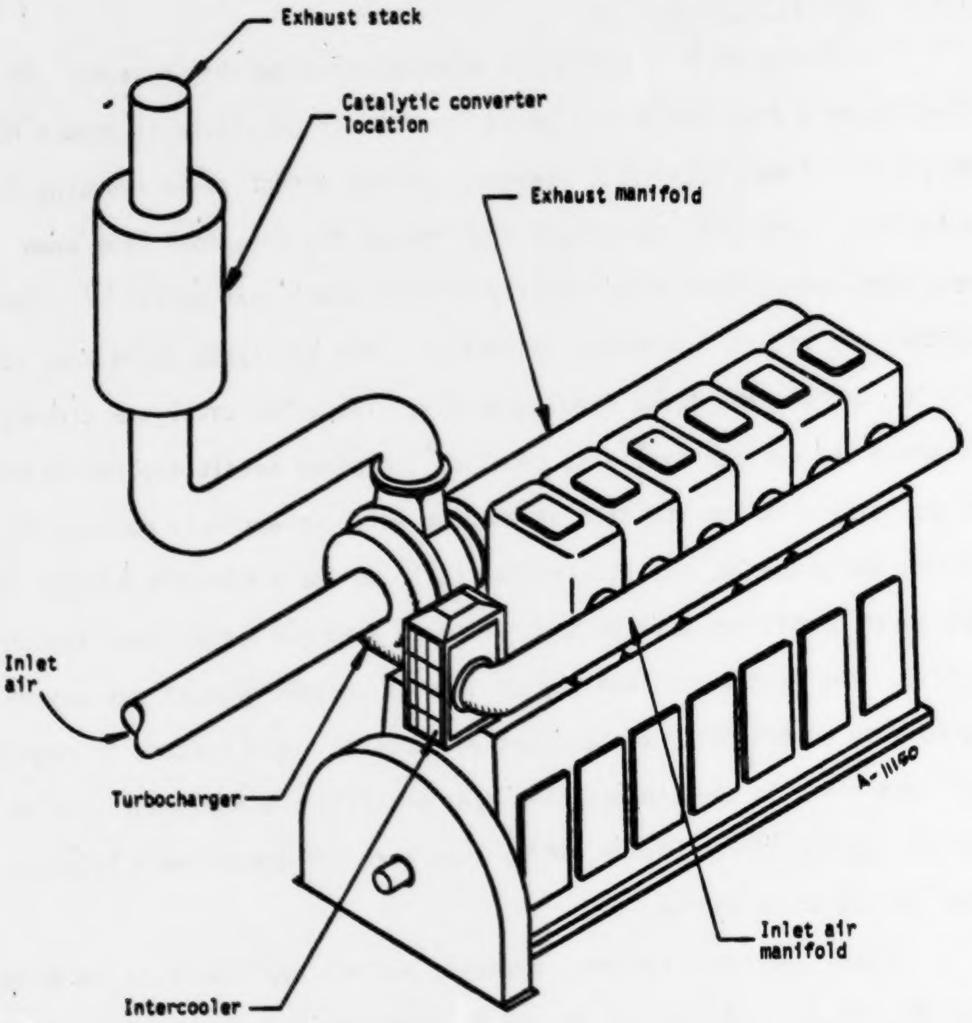


Figure 4-47. Possible location of a catalytic muffler on a large-bore engine.

are typically much greater than stoichiometric. Consequently, other approaches which can selectively reduce $NO_x$ emissions in the presence of excess oxygen must be considered.

One approach is to inject ammonia, hydrogen, CO, or natural gas into the exhaust to create the required reducing atmosphere(132,133). Ammonia appears to work best because it has an affinity for the NO rather than the excess oxygen. That is, one needs to add only slightly more ammonia than would be required for stoichiometric reaction with the NO, whereas with the other three substances, additional quantities are required to deplete the excess oxygen in the exhaust. Because of the high cost (see Section 8.2 in Chapter 8), even with ammonia, no engine manufacturer is investigating the reduction of $NO_x$ in IC engines using ammonia(134).

One research laboratory has experimented with a combination converter scrubber exhaust treatment system for a two-stroke diesel engine commonly used in buses(135). $NO_x$ scrubbers are actually $NO_2$ scrubbers and are ineffective for NO. Since NO is the primary constituent of $NO_x$ in the exhaust, scrubbers cannot be used on engine exhaust unless the NO is converted to $NO_2$ first. Therefore, this laboratory converted the NO to $NO_2$ by passing the exhaust through a catalytic converter which also oxidized the CO, HC, and oxygenated HC (odorous gases) to $CO_2$ and water. Since the exhaust gas temperature was below the 700°F required by the converter for proper operation, the exhaust was first passed through a heat exchanger to raise its temperature. Finally the oxidized exhaust gas entered a molten carbonate scrubber where the $NO_x$ was reduced by approximately 50 percent (reduction from 17.35 g/hphr to 9.2 g/hp-hr). This reduction fell short of the desired 70-percent reduction (to meet a 5 g/hp-hr standard). In addition, the cost of the system was judged unacceptable due to the high price of the platinum catalyst ($15/lb) used. With a system requirement of 60 lb of catalyst, the platinum catalyst, alone, would cost approximately $900.

In conclusion, then, catalytic reduction of $NO_x$ has not, as of the present, been demonstrated as a viable control technology for large-bore stationary reciprocating engines.

### 4.4.10 Combinations of Controls

Several manufacturers have conducted experiments with more than one control system at a time. The results of these tests are presented by fuel in Figure 4-48. Figure 4-48(a) indicates that the controls are nearly additive for diesels. Although a greater reduction is achieved on gas and dual fuel engines by the combination of controls than by any one of the individual component techniques when used alone, the reduction is less than the sum of the effects from each technique.

Thus, for the large-bore diesel engine shown in Figure 4-48(a) the maximum $NO_x$ reduction for a single control (retard) is 1.7 g/hp-hr. When all the controls tested (retard, reduced inlet manifold air temperature, increased air-to-fuel ratio, and water induction) were applied simultaneously, $NO_x$ was reduced 3.0 g/hp-hr. This is shown on the figure as an uninterrupted downward arrow. For comparison, to the left of this, is a multiple arrow line that represents the depiction in series of all the separate control effects, as they were individually measured. The difference between the length of these two lines is a measure of the relationship between the additive effects of the controls when applied simultaneously and the sum of their individual contributions. As noted earlier, for this case the controls do appear to be additive.

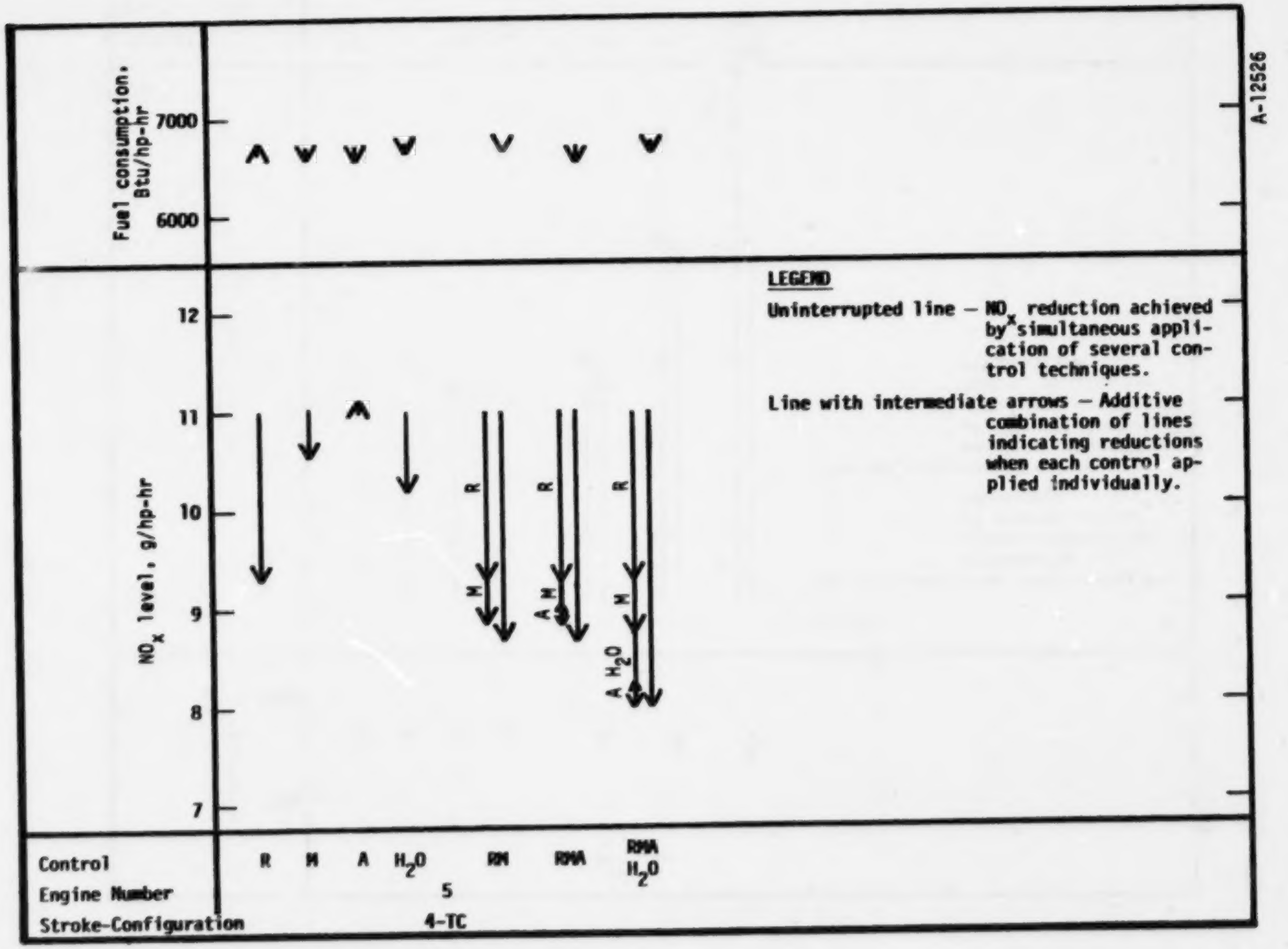


Figure 4-48(a). Additive effects of controls for a large-bore diesel engine.

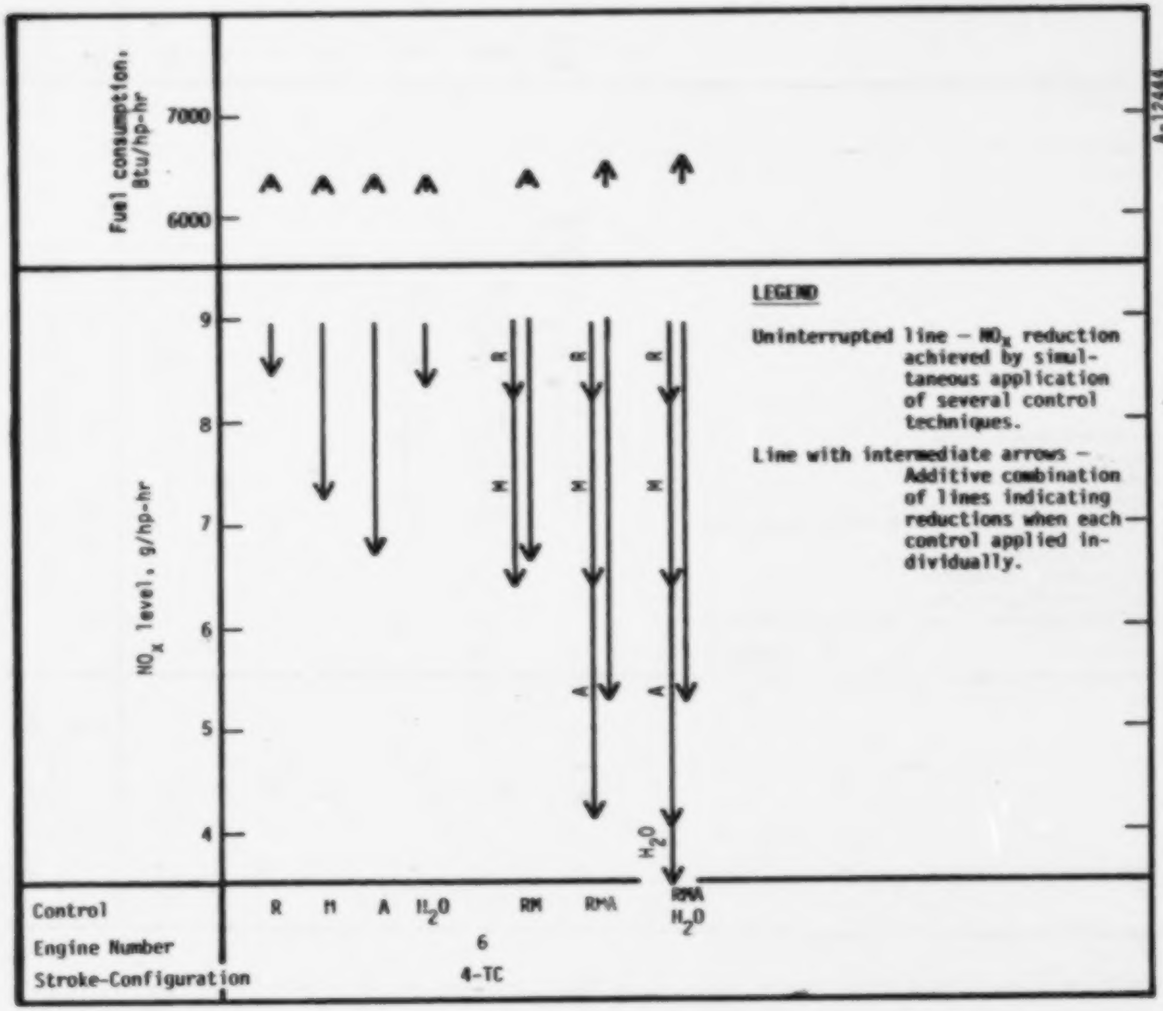


Figure 4-48(b). Additive effects of controls for a large-bore dual-fuel engine.

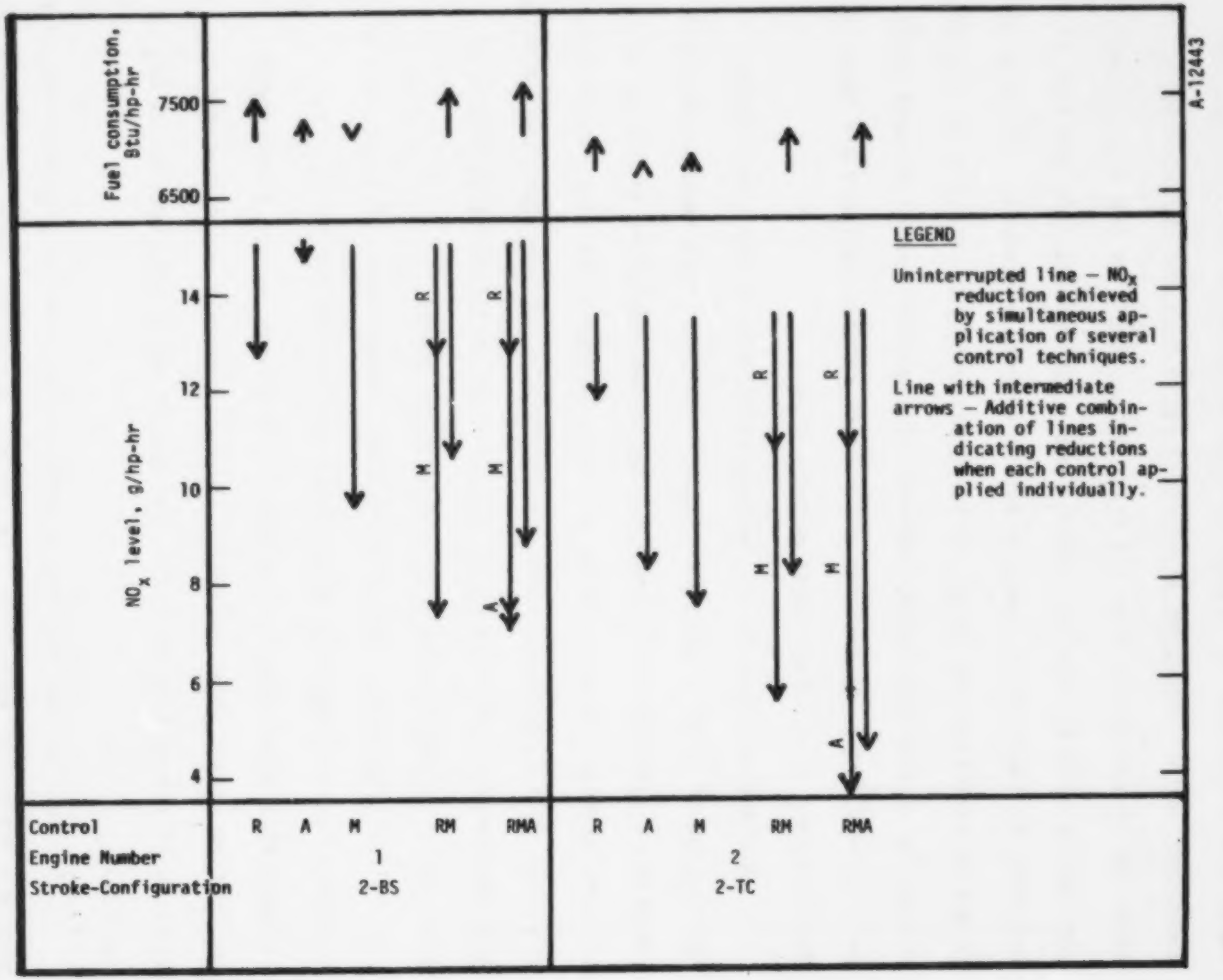


Figure 4-48(c). Additive effects of controls for large-bore gas engines.

Figure 4-48(b) shows that the combination of retard and manifold air temperature reduction is nearly additive in dual-fuel engines, but the additional use of increased air-to-fuel ratio does not decrease $NO_x$ as much below the levels obtained by the first two controls as one would expect from the results with air-to-fuel ratio alone. Furthermore, the addition of water induction to these three controls has no effect, whatsoever. If moderate $NO_x$ reduction were required (e.g., 25 to 30 percent) for this engine, one would change the air-to-fuel ratio; however, if a greater reduction were necessary one would use the combined controls of retard, manifold air temperature decrease, and air-to-fuel ratio modification.

Gas engines also do not respond additively to the simultaneous application of several controls (Figure 4-48(c)). Here reduced manifold air temperature decreased emissions from the blower-scavenged engine by 36 percent, whereas the application of reduced temperature plus retard reduced emissions only 29 percent. Only the combination of the above two techniques with increased air-to-fuel ratio could reduce emissions below the level obtained by reduced manifold air temperature alone.

The turbocharged engine reached its lowest level (7.4 g/hp-hr) with an individual control when the air-to-fuel ratio modification was used. Although this engine was controlled to 4.5 g/hp-hr by the simultaneous application of all three controls, if these controls would have had an additive effect on the emissions, the engine would have emitted no $NO_x$ whatsoever.

All but two combinations of controls resulted in a greater reduction than any individual technique. In one gas engine the combinations without increased air-to-fuel ratio were not as effective as increased air-to-fuel ratio by itself. Recent tests on installed gas pipeline engines have shown that increased air-to-fuel ratio is the most significant source of $NO_x$ reductions from large-bore gas engines(136). In the other case, the combinations without increased air-to-fuel ratio were not as effective as increased aftercooling by itself.

### 4.4.11 Summary of $NO_x$ Reductions

This subsection summarizes the data presented in Sections 4.4.1 through 4.4.10 for control techniques that have been shown to be technically viable and effective in reducing $NO_x$ emissions. These techniques include: derate, retard, air-to-fuel changes, manifold air temperature reductions, exhaust gas recirculation (EGR), and combinations of these controls. With the exception of EGR, all of these techniques consist of a change in an engine operating condition.

Although water induction has been shown effective in reducing $NO_x$ emissions, no data have been summarized, since this technique has serious technical limitations (see Section 4.4.7). Furthermore, modifications to combustion chamber geometries (Section 4.4.8) and the application of catalytic converters (Section 4.4.9) have been excluded from this summary since only limited data for large-bore engines are available for these control approaches. Nevertheless, both of these approaches have been used successfully on smaller bore, mobile engines. In general, however, these techniques will require considerable development before they can be considered technically viable considering costs. Turbocharging (see Section 4.4.4) has not been listed as a control approach since most large-bore engines are already turbocharged, and engine users are showing increasing preference for turbocharged units because of their lower initial costs(137).

A summary of the $NO_x$ reductions achieved by the candidate control techniques discussed above is presented on Figure 4-49. The arrows on these figures begin at the uncontrolled $NO_x$ level and end at the controlled level. Dashed arrows indicate the same reduction after correction to standard ambient condition (85°F, 75 grains/$ft^3$ humidity) (see Section 4.2.1.) The changes in fuel consumption after application of $NO_x$ controls are also shown in Figure 4-49. Figure 4-50 shows uncontrolled $NO_x$ levels and the lowest controlled level for each manufacturer of diesel, dual-fuel, and gas engines. Note in Figure 4-50 that some of the engines show only uncontrolled emissions since no control techniques had been applied.

A $NO_x$ control data summary appears in Table 4-13 based on Figures 4-65 through 4-70. This table also shows uncontrolled levels and the lowest controlled level for each kind of control technique as applied to each engine type and fuel combination. These data are summarized in a different way in Table 4-14.

- Uncontrolled levels for dual-fuel engines are generally lower than for diesels, which are generally lower than for gas engines
- Diesel, dual-fuel, and 2-TC gas engines more consistently achieve lower $NO_x$ levels than do 4-TC gas engines. For example, at least one turbocharged diesel, dual-fuel, or two-stroke gas engine is capable of reaching a $NO_x$ level of 6 g/hp-hr. In comparison, the lowest $NO_x$ level (corrected for ambients) demonstrated by any for which data are available is 9 g/hp-hr.
- In general, the data show that two-stroke turbocharged gas and diesel engines have reached lower levels than blower-scavenged or naturally aspirated engines. No trend is apparent for the same comparisons of four-stroke diesel and gas engines.

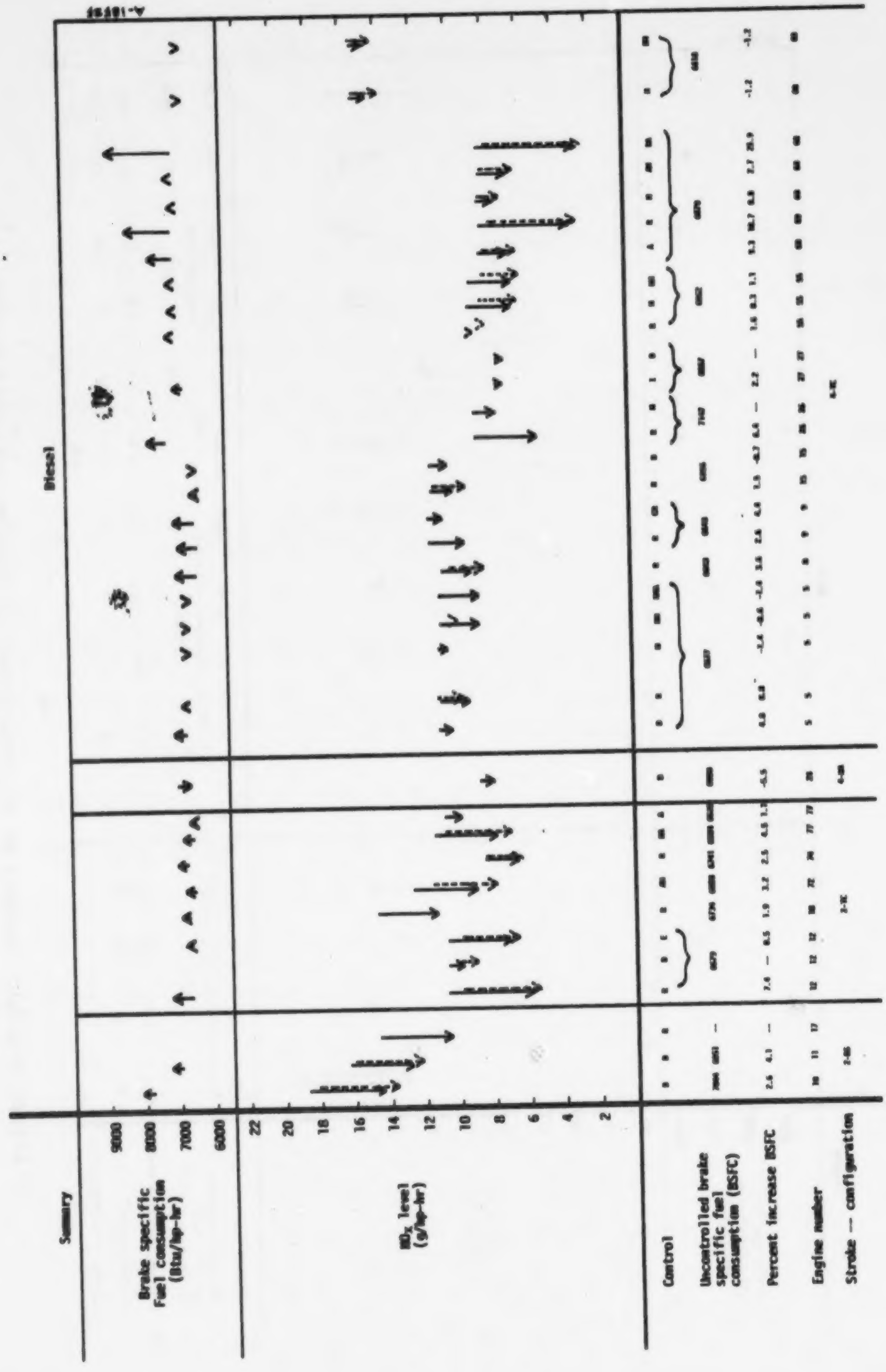
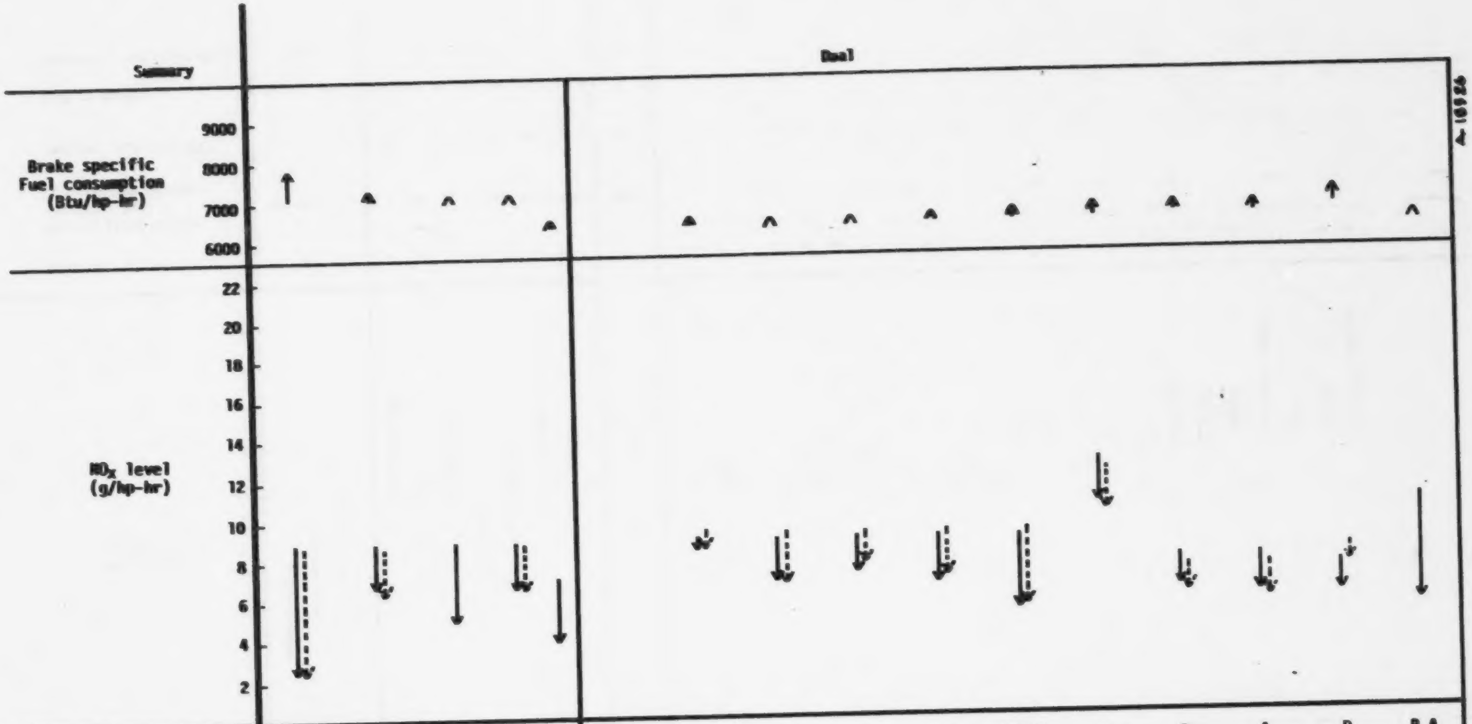


Figure 4-49(a). Summary of NO$_x$ reductions achieved by diesel engines.



| Control | R | A | M | E | R,A | R | A | M | R,M | R,M,A | D | R | A | D | R,A |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Uncontrolled brake specific fuel consumption (BSFC) | 7040 | | | | 6480 | 6390 | | | | | 6400 | 6464 | | 6570 | 6208 |
| Percent increase BSFC | 10.1 | 2.5 | 0.5 | 0.8 | 2.1 | 1.1 | 0.8 | 0.6 | 1.6 | 3.0 | 3.1 | 2.9 | 2.8 | 6.1 | 0.9 |
| Engine number | 13 | 13 | 13 | 13 | 78 | 6 | 6 | 6 | 6 | 6 | 7 | 16 | 16 | 61 | 76 |
| Stroke — configuration | 2-TC | | | | | 4-TC | | | | | | | | | |

Figure 4-49(b). Summary of $NO_x$ reductions achieved by dual-fuel engines.

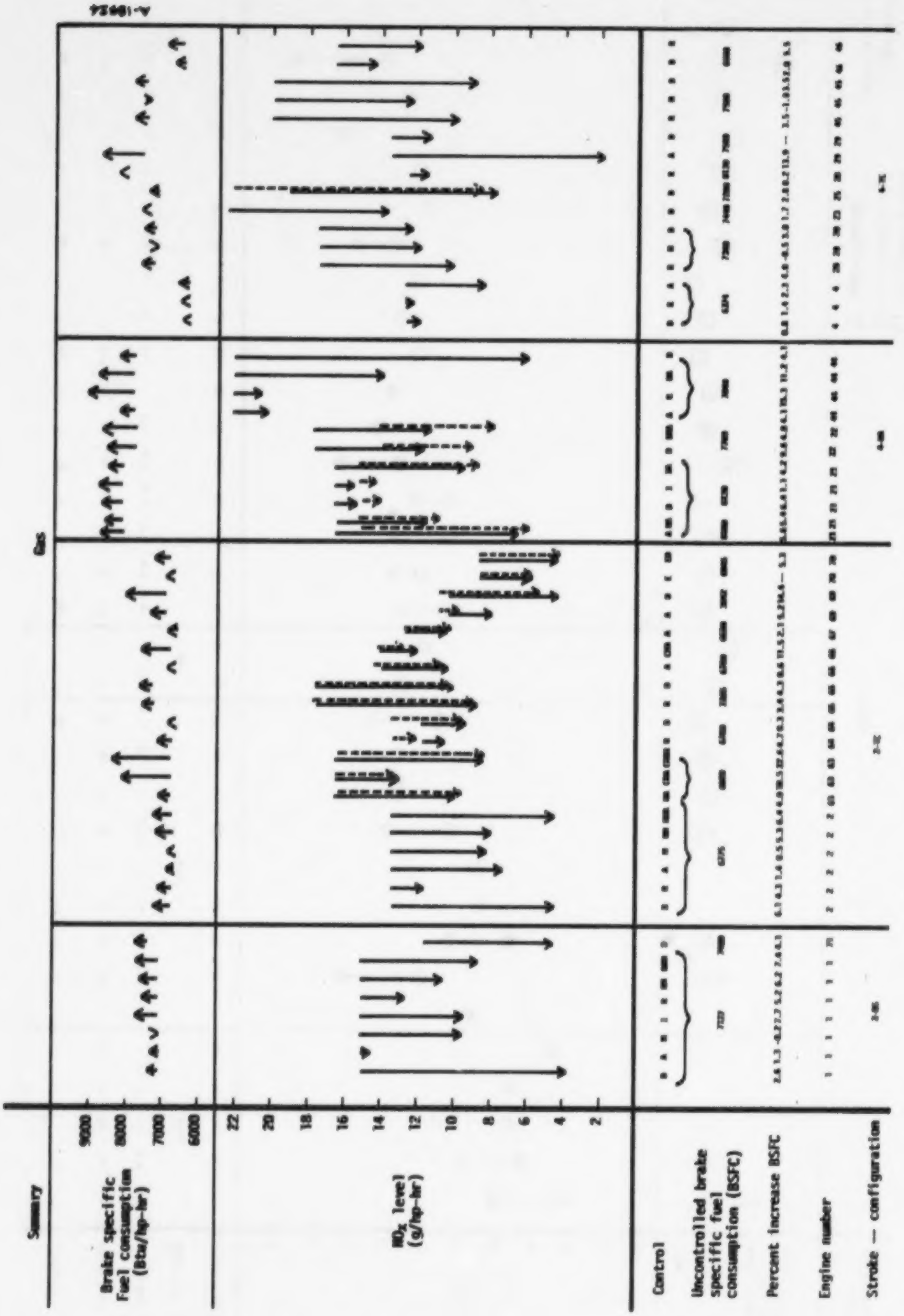


Figure 4-49(c). Summary of $NO_x$ reductions achieved by gas engines.

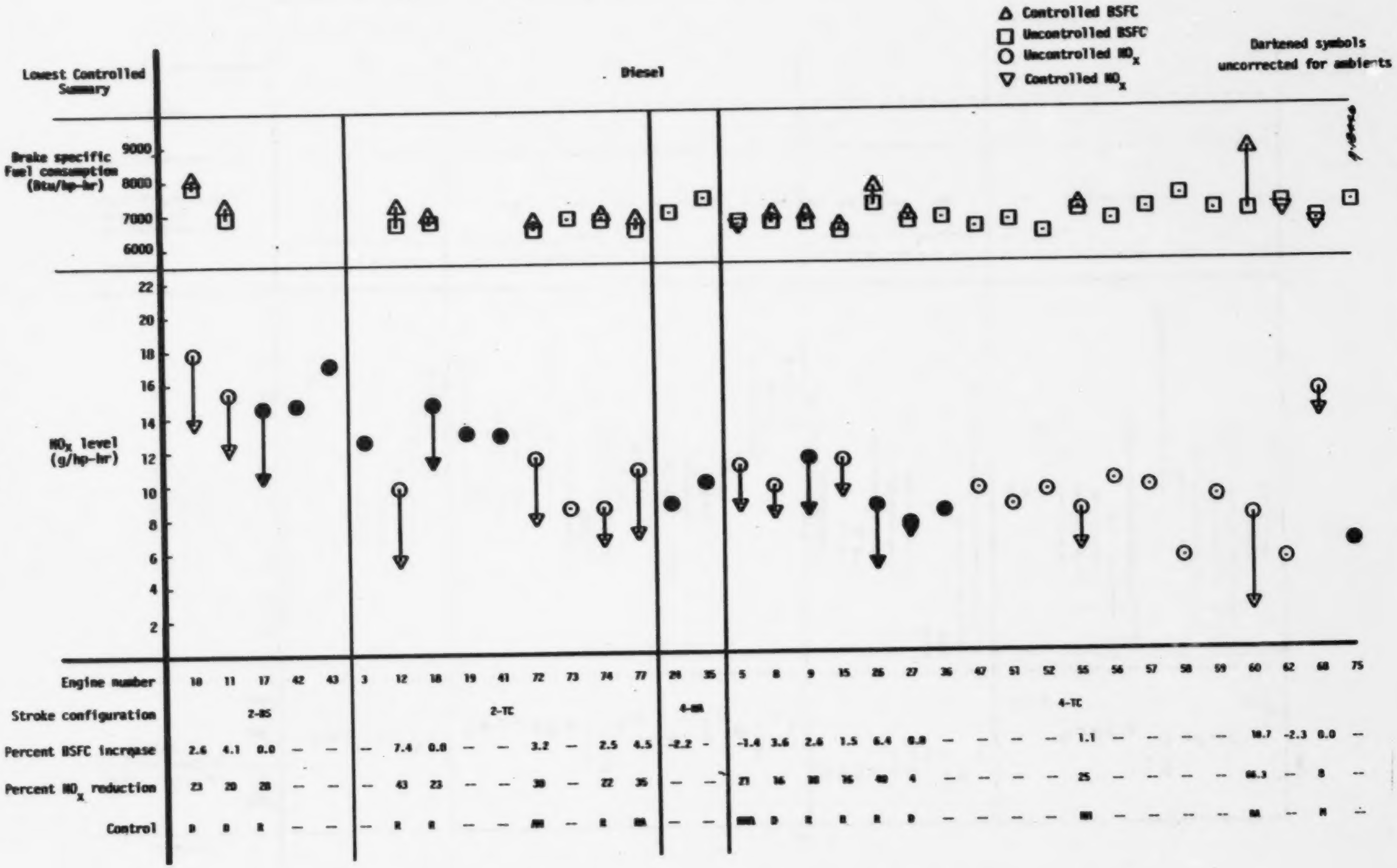
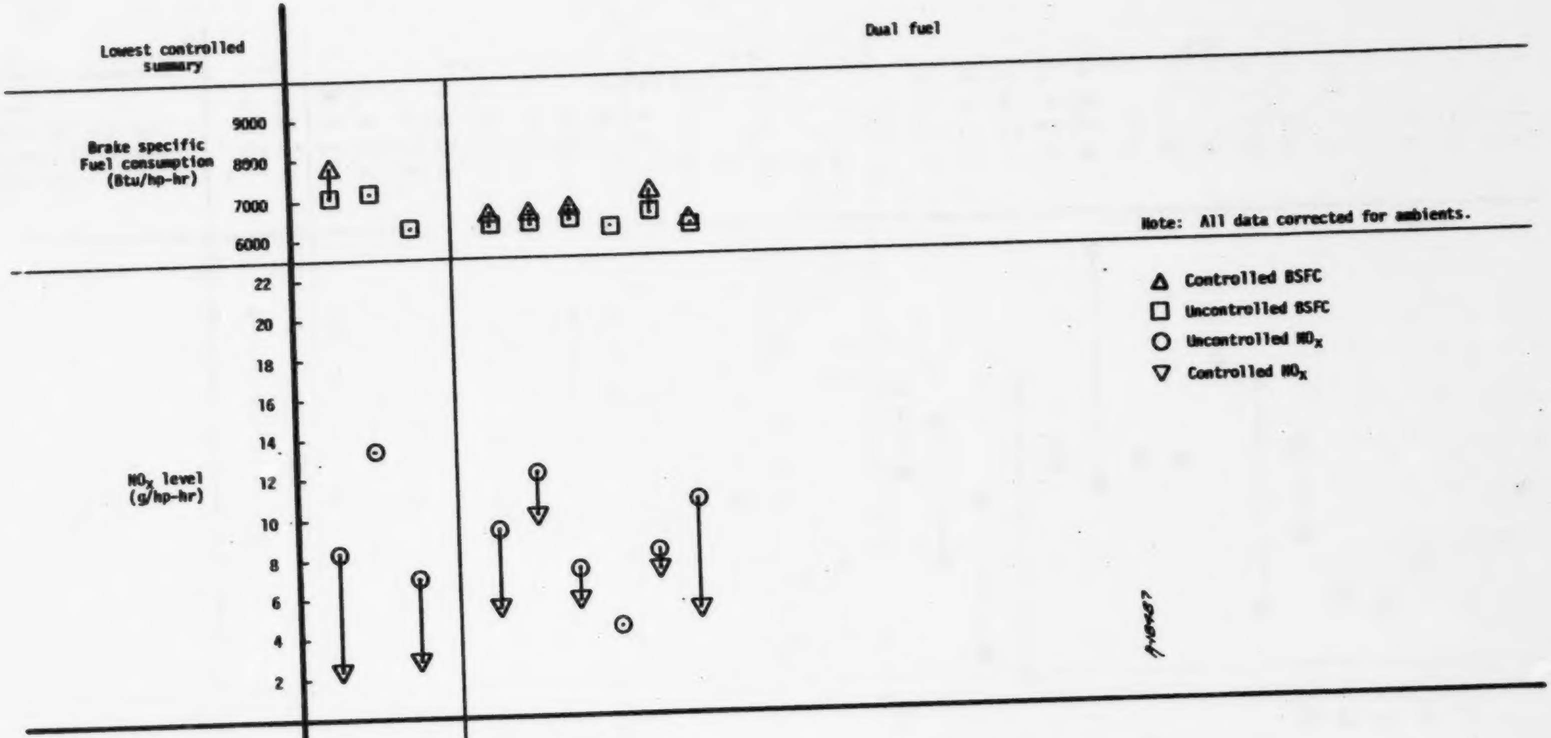
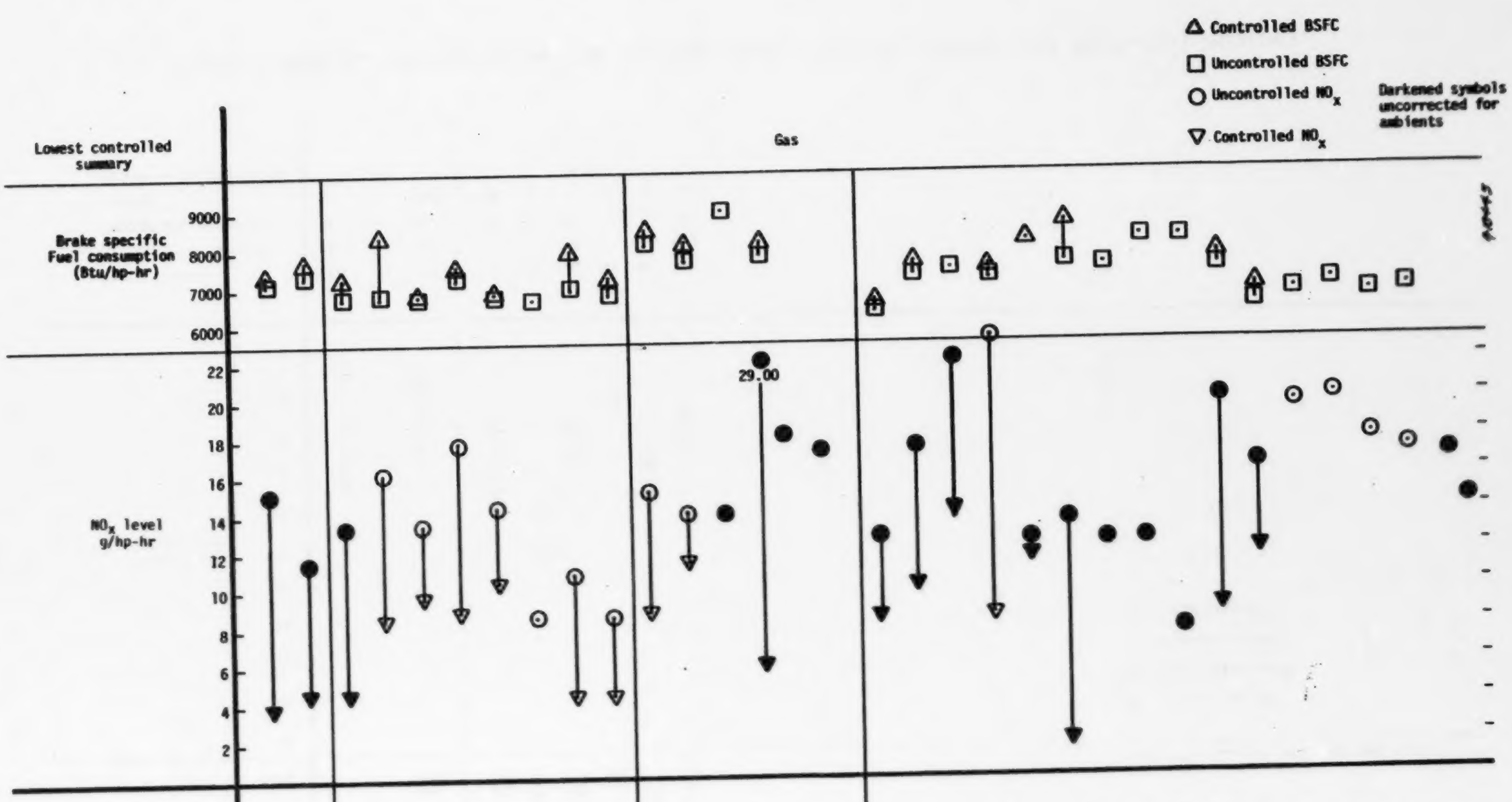


Figure 4-50(a). Uncontrolled and maximum controlled NO$_x$ levels for diesel engines.



| Engine Number | 13 | 14 | 78 | 6 | 7 | 16 | 50 | 61 | 76 |
|---|---|---|---|---|---|---|---|---|---|
| Percent BSFC increase | 10.1 | -- | 3.1 | 3.0 | 3.1 | 2.8 | -- | 6.1 | 0.9 |
| Percent $NO_x$ reduction | 74 | -- | 56 | 41 | 16 | 22 | -- | 9 | 49 |
| Control | R | -- | R,A | RM | D | A | -- | D | R,A |
| Stroke -- configuration | 2-TC | | | 4-TC | | | | | |

Figure 4-50(b). Uncontrolled and maximum controlled $NO_x$ levels for dual-fuel engines.



| Engine number | 1 | 71 | 2 | 63 | 64 | 65 | 66 | 67 | 69 | 70 |
|---|---|---|---|---|---|---|---|---|---|---|
| Percent BSFC increase | 2.6 | 4.1 | 6.4 | 22.6 | 0.3 | 3.4 | 0.6 | 0.0 | 14.4 | 5.3 |
| Percent $NO_x$ reduction | 75 | 60 | 66 | 48 | 27 | 49 | 24 | -- | 49 | 49 |
| Control | D | D | NMA | CR,A,R | D | R | A | A | D | EE,D |
| Stroke -- configuration | 2-BS | | 2-TC | | | | | | | |

| Engine number | 21 | 22 | 39 | 44 | 79 | 80 |
|---|---|---|---|---|---|---|
| Percent BSFC increase | 4.2 | 4.8 | -- | 4.1 | -- | -- |
| Percent $NO_x$ reduction | 42 | 43 | -- | 79 | -- | -- |
| Control | EE,A | D,R,A | -- | D | -- | -- |
| Stroke -- configuration | 4-NA | | | | | |

| Engine number | 4 | 20 | 23 | 25 | 28 | 29 | 37 | 38 | 40 | 45 | 46 | 48 | 49 | 53 | 54 | 81 | 82 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Percent BSFC increase | 2.3 | 4.0 | 1.7 | 6.4 | 0.2 | 13.9 | -- | -- | -- | 3.5 | 5.5 | -- | -- | -- | -- | -- | -- |
| Percent $NO_x$ reduction | 31 | 42 | 38 | 61 | 6 | 88 | -- | -- | -- | 55 | 27 | -- | -- | -- | -- | -- | |
| Control | A | R | D* | D | R | A | -- | -- | -- | D | R | -- | -- | -- | -- | -- | -- |
| Stroke -- configuration | 4-TC | | | | | | | | | | | | | | | | |

Figure 4-50(c). Uncontrolled and maximum controlled $NO_x$ levels for gas engines.

TABLE 4-13. SUMMARY OF LOWEST $NO_x$ LEVELS (g/hr-hr) FROM CONTROLLED LARGE-BORE ENGINES

T-655

| Fuel | Diesel | | | | Dual Fuel | | Gas | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| Strokes/Cycle | 2 | | 4 | | 2 | 4 | 2 | | 4 | |
| Air Charging | BS | TC | NA | TC | TC | TC | BS | TC | NA | TC |
| Uncontrolled | 14.6-17.8 | 8.5-14.7 | 8.7-10.0 | 5.2-15.2 | 6.9-13.4 | 4.4-12.1 | 11.5-15.1 | 8.6-17.7 | 14.0-29.0 | 7.8-22.3 |
| Derate, D | 13.8 | 9.0 | 10.7 | 7.1 | -- | 7.3 | 3.8 | 4.4 | 6.0 | 8.6 |
| Retard, R | 10.6 | 5.6 | -- | 3.0 | 2.5 | 8.5 | 12.8 | 11.8 | 27.9 | 10.0 |
| A/F, A | -- | -- | -- | 6.3 | 6.3 | 5.8 | 14.7 | 7.4 | 20.5 | 2.0 |
| MAT, M | -- | -- | -- | 6.2 | 5.0 | 7.7 | 9.6 | 8.3 | -- | 12.0 |
| Internal EGR, IE | -- | -- | -- | 7.1 | -- | -- | 9.5 | -- | 14.4 | -- |
| External EGR, EE | -- | 6.6 | -- | -- | 6.5 | -- | -- | 5.8 | -- | -- |
| R+M | -- | -- | -- | 6.2 | -- | 7.0 | 10.6 | 8.1 | -- | -- |
| R+A | -- | 7.0 | -- | 2.6 | -- | -- | -- | 9.8 | 10.9 | -- |
| D+M | -- | -- | -- | 14.2 | -- | -- | -- | -- | -- | -- |
| A+M | -- | 8.0 | -- | 6.3 | -- | -- | -- | -- | -- | -- |
| R+M+A | -- | -- | -- | 8.7 | -- | 5.5 | 8.7 | 4.6 | -- | -- |
| Best Control | 10.6 | 5.6 | 10.7 | 2.5 | 2.5 | 3.8 | 4.4 | 4.4 | 6.0 | 2.0 |

TABLE 4-14. $NO_x$ EMISSIONS FROM VARIOUS GROUPINGS OF ENGINES

| Group | Emissions, g/hp-hr | |
|---|---|---|
| | Uncontrolled | Controlled |
| Air charging | | |
| Turbocharged | 4.4-22.3 | 2.0-5.6 |
| Nonturbocharged | 8.7-29.0 | 3.8-10.7 |
| Strokes/cycle | | |
| 2 | 6.9-17.8 | 2.5-10.6 |
| 4 | 4.4-29.0 | 2.0-10.7 |
| Fuel | | |
| Diesel | 5.2-17.8 | 2.6-10.7 |
| Dual Fuel | 4.4-13.4 | 2.5-5.5 |
| Gas | 7.8-29.0 | 2.0-6.0 |

Since emission levels from uncontrolled engines vary so widely, the environmental impact depends more on the percentage $NO_x$ reduction it produces than on absolute $NO_x$ level it achieves. Therefore, the data shown in Figure 4-49 have been recost in terms of percentage $NO_x$ reductions and summarized in Table 4-15. A number of techniques, either alone or in combination, are capable of achieving $NO_x$ reductions of 20 to 40 percent with moderate increases in fuel consumption (less than 10 percent). In general, retard, air-to-fuel ratio changes, or combinations of these two techniques achieved greater than 40 percent $NO_x$ reductions.

In addition to these techniques, combinations of retard and manifold air cooling produced similar results for gas and dual-fuel engines. As shown in Figure 4-49, a number of turbocharged diesel, dual-fuel and gas engines achieved 20 to 40 percent $NO_x$ reductions with less than a 10-percent increase in fuel consumption. Thus, several techniques can effectively reduce $NO_x$ emissions from diesel large-bore. With the exception of controls applied to dual fuel engines, however, maximum $NO_x$ reductions are accompanied by large increases in fuel consumption.

Figure 4-51 presents an overview of the fuel penalties associated with the various $NO_x$ control techniques. Based on the data shown in Figure 4-51, large-bore diesel engines typically suffer a fuel consumption increase of somewhat less than 1 percent for each g/hp-hr of $NO_x$ reduction. The major deviations from this average figure are manifold air temperature reduction, which in some cases actually causes fuel consumption to decrease, and derating which is accompanied by higher than average fuel penalties. Retard, whether used alone or in combination A/F ratio changes, causes the largest fuel penalties, but also achieves the lowest $NO_x$ levels.

TABLE 4-15. ALTERNATIVE CONTROL TECHNIQUES -- SUMMARY OF SELECTED CONTROLS

| % $NO_x$ Reduction | Diesel | | Dual Fuel | | Gas | |
|---|---|---|---|---|---|---|
| | Control | ΔBSFC, % | Control | ΔBSFC, % | Control | ΔBSFC, % |
| $\leq 20$ | A, M, IE, RMA, D, R, EE, RM, RA, AM | 1 to 8 | EE, D, R, A, M, RM, RA, RMA | 0 to 4 | D, R, A, M, IE, EE, RM, RA, RMA | 0 to 5 |
| 20 and $\leq 40$ | D, EE, RM, AM, R, RA | 1 to 16 | D, A, M, RM, RMA, R, RA | 1 to 7 | R, M, IE, EE, RM, D, A, RA, RMA | 0 to 8 |
| >40 | R, RA | 8 to 26 | R, RA | 2 to 5 | D, A, RA, RMA | 5 to 12 |
| Maximum reduction (%) | RA (65) | 26 | R (73) | 3 | D (90) | 12 |

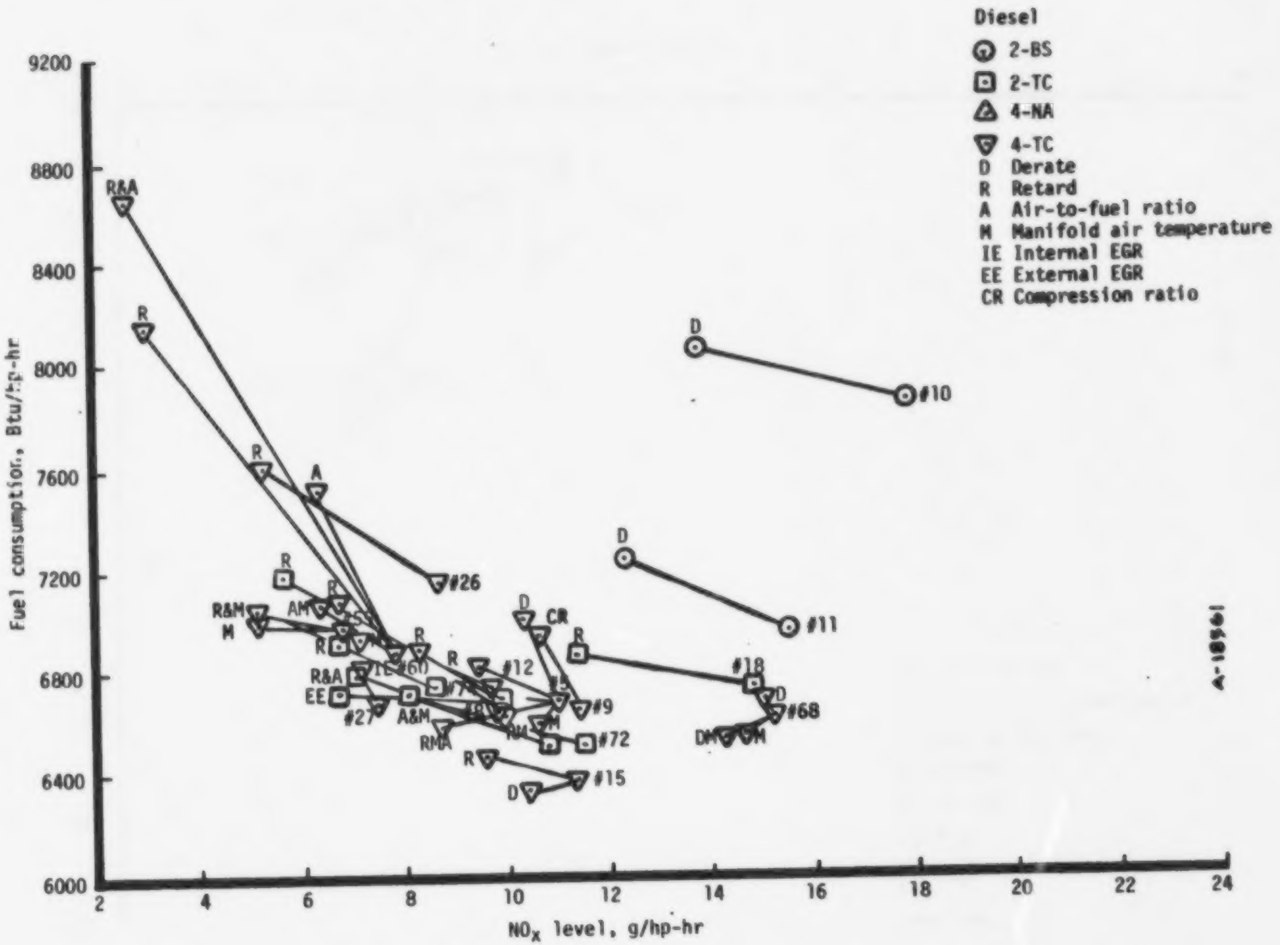


Figure 4-51(a). Fuel consumption versus controlled $NO_x$ level for diesel engines.

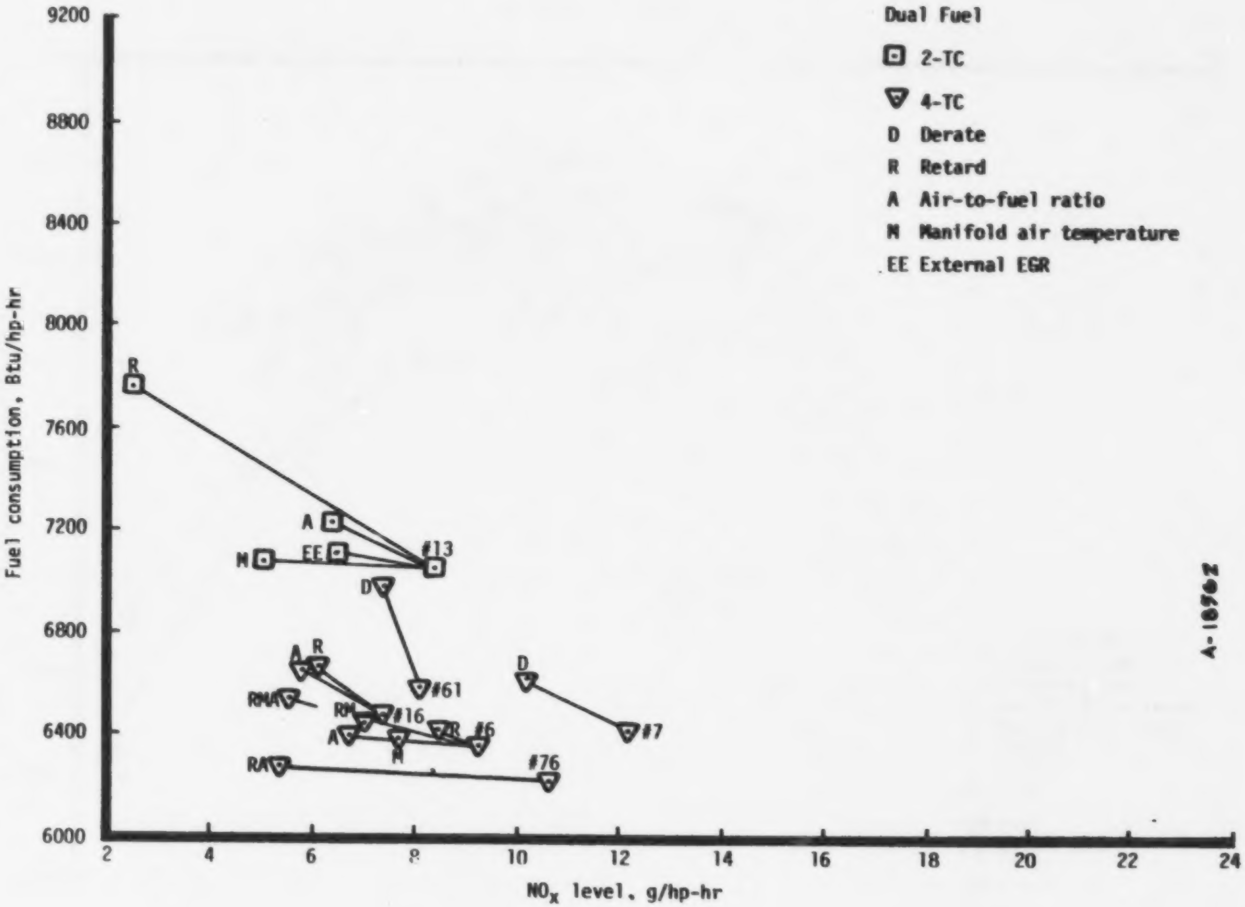


Figure 4-51(b). Fuel consumption versus controlled $NO_x$ level for dual-fuel engines.

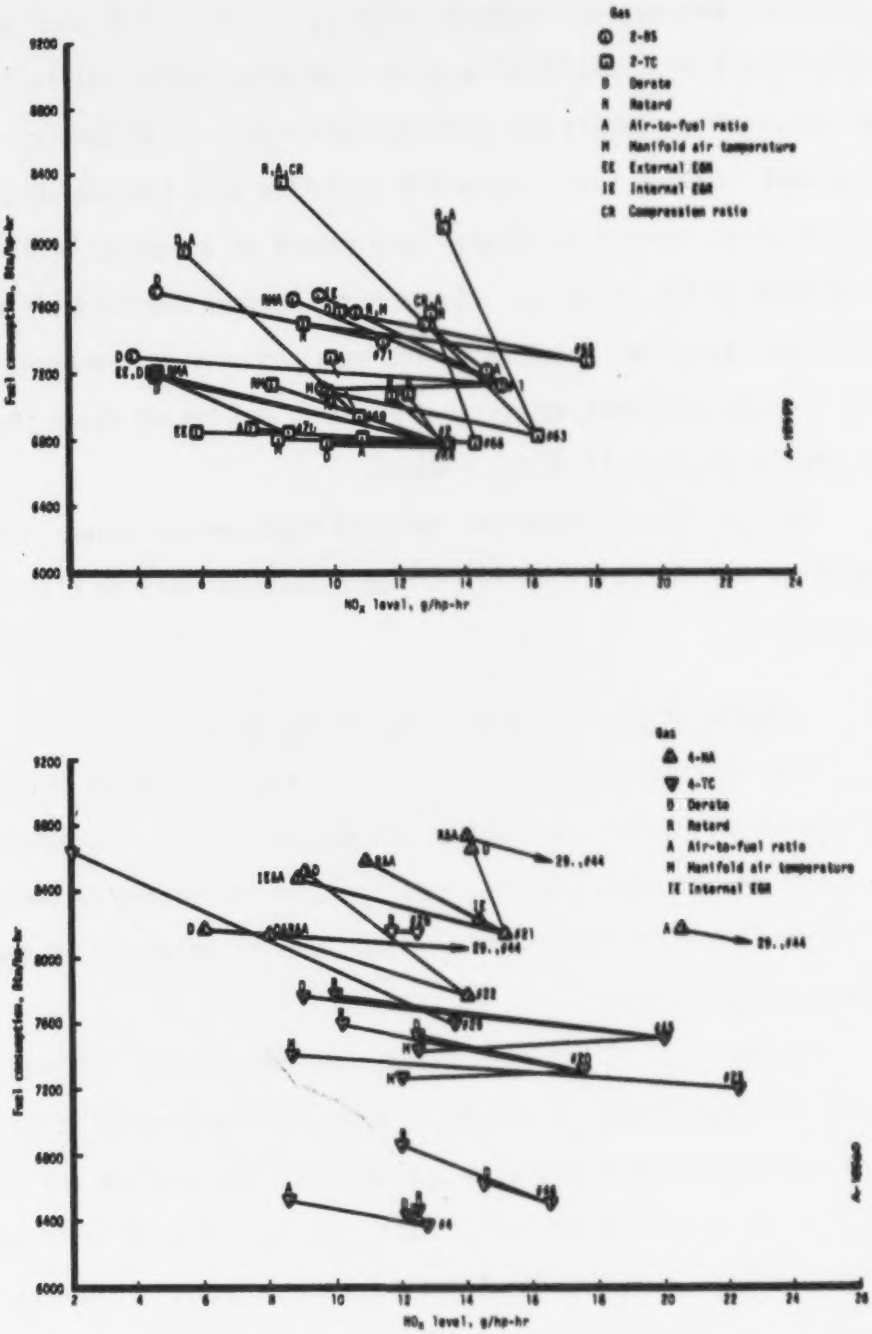


Figure 4-51(c). Fuel consumption versus controlled NOx level for gas engines.

Fuel consumption increases slightly more than 1 percent per g/hp-hr of $NO_x$ reduction in dual-fuel engines. Derating again results in the worst fuel consumption penalty per unit $NO_x$ reduction. Large-bore gas engines show the widest range of fuel consumption variation with the use of controls. Fuel penalties range from slight improvements to a maximum of 1.8 percent per g/hp-hr of $NO_x$ reduction. In general, reduced manifold air temperature causes the least fuel consumption increase per unit $NO_x$ removed. Air-to-fuel ratio changes and combinations of A/F change and retard cause the largest fuel penalties per unit of $NO_x$ reduced.

The techniques summarized here and their effectiveness, their fuel penalties, and the time required to implement them will be discussed further in Chapter 6.

### 4.4.12 Effects of $NO_x$ Controls on HC, CO and Smoke

This section discusses the effect of the application of the $NO_x$ controls on the criteria pollutants HC, CO, and smoke. The purpose of this discussion is to insure that $NO_x$ emissions can be reduced by the techniques discussed above without unacceptable increases in emissions of these other criteria pollutants.

The effect of $NO_x$ control systems on hydrocarbons is shown graphically by fuel in Figure 4-52. It should be noted that only about 10 to 25 percent of the hydrocarbon emissions from gas and dual-fuel engines are reactive (i.e., criteria hydrocarbon pollutants), the remaining 75 to 90 percent being unburned methane (see Appendix C.4 and Reference 138). In contrast, over 90 percent of the total hydrocarbon emissions from diesel engines are reactive(139), since diesel fuel is composed of higher order hydrocarbon molecules (i.e., there are many hydrogen and carbon atoms in each molecule).

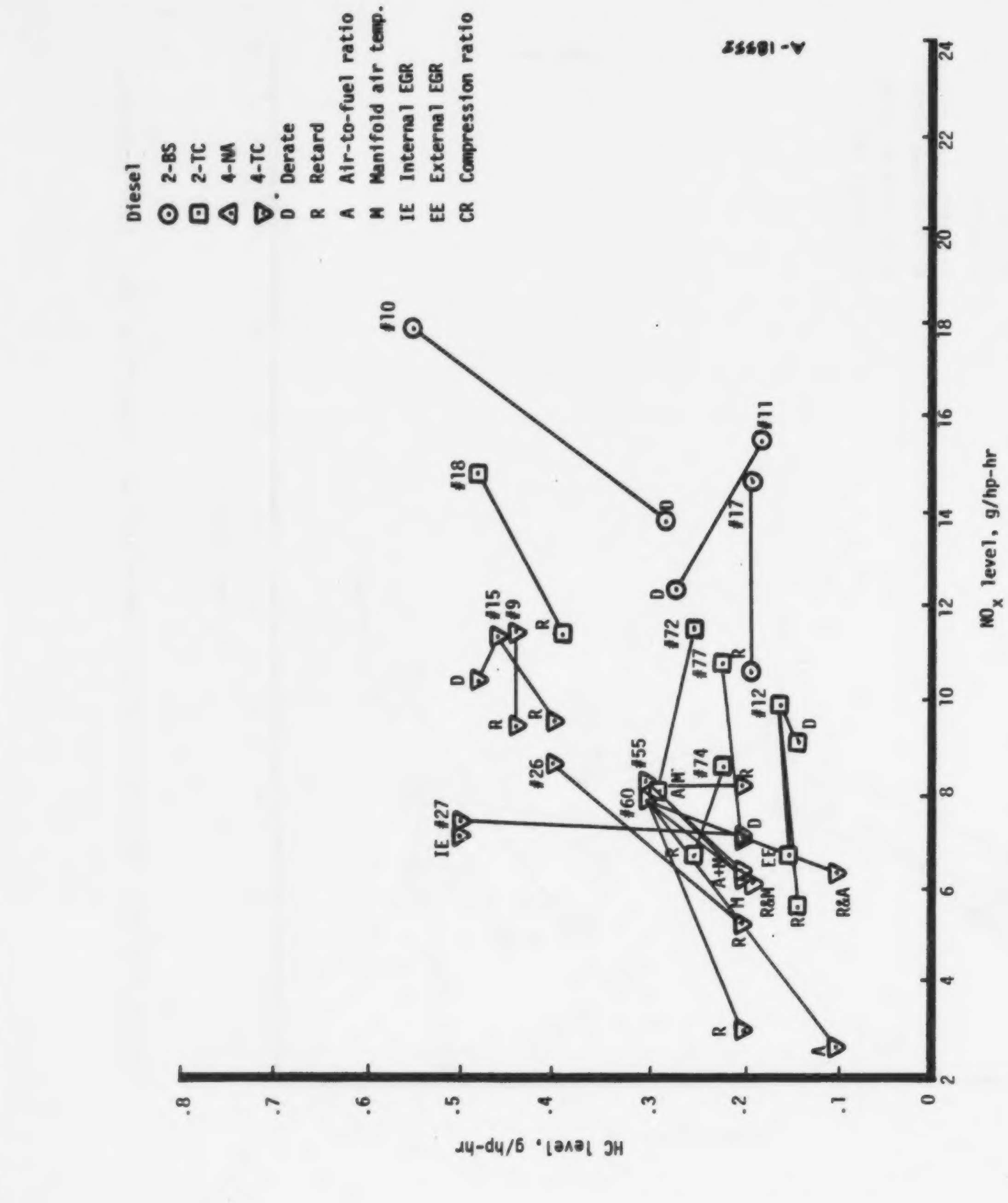


Figure 4-52(a). HC levels versus controlled $NO_x$ levels for diesel engines.

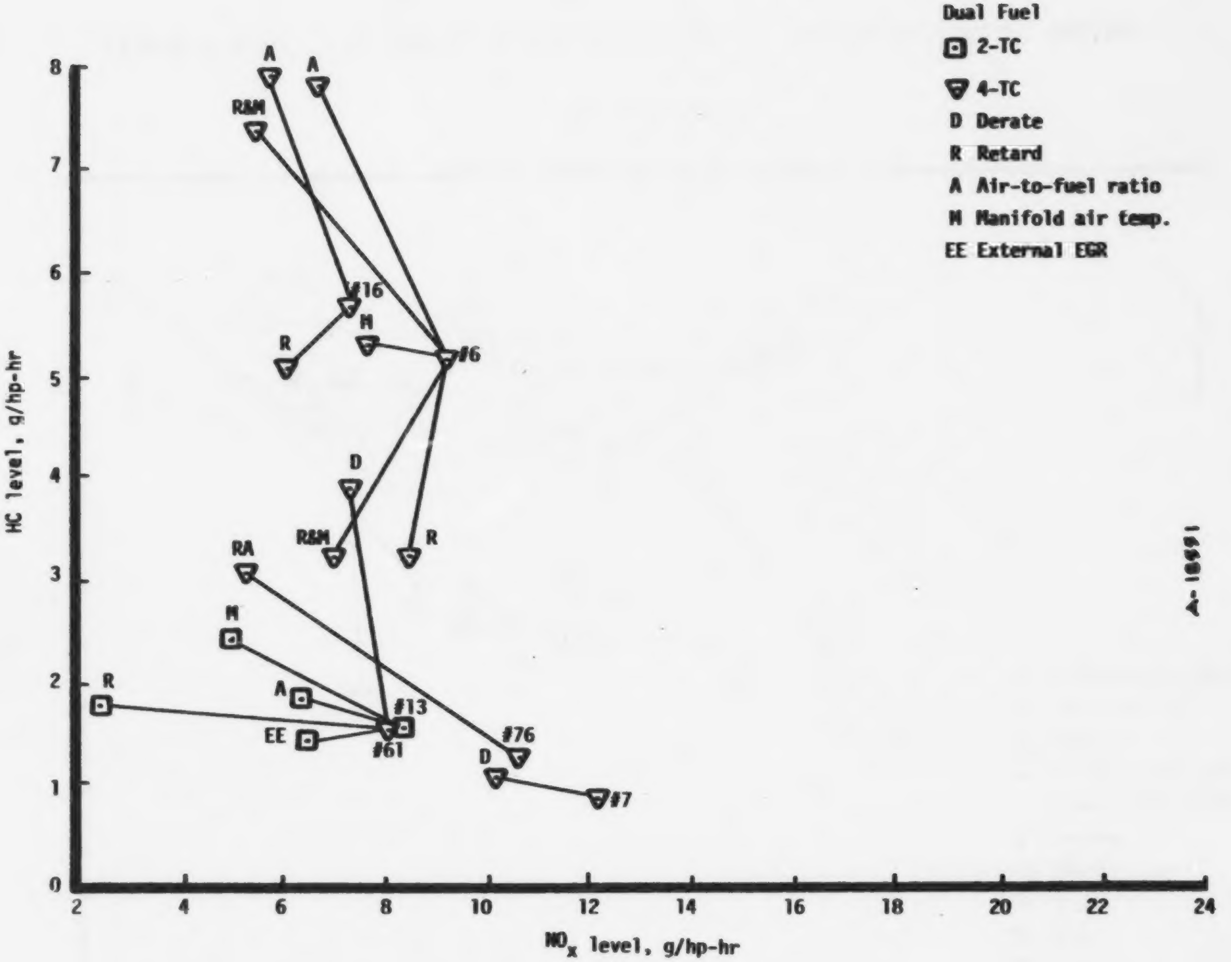


Figure 4-52(b). HC levels versus controlled $NO_x$ levels for dual-fuel engines.

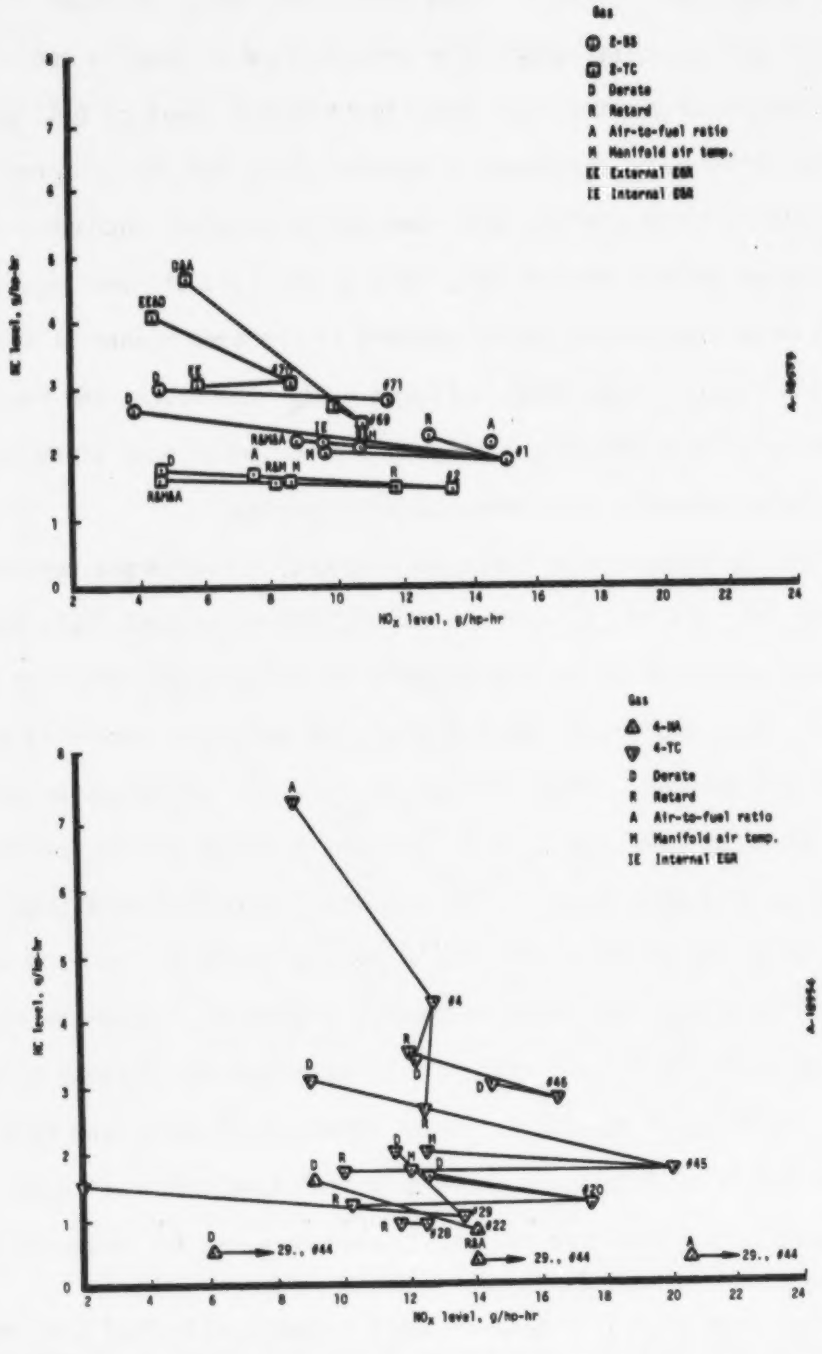


Figure 4-52(c). HC levels versus controlled $NO_x$ levels for gas engines.

Therefore, the reactive emissions from most engines range between 0.1 and 0.5 g/hp-hr, independent of fuel. These values are nearly an order of magnitude lower than corresponding numbers for uncontrolled automobile engines and within a factor of two from the ultimate statutory limit of 0.41 $gHC_t$/mile.[8/]

Furthermore, as discussed in Appendix C.4, the ratio of reactive hydrocarbons to total hydrocarbons remains, in general, unchanged after the application of derate, air-to-fuel, retard, manifold air cooling, and EGR. Thus, reactive hydrocarbon levels respond in the same manner as the total hydrocarbon levels illustrated in Figure 4-52. Therefore, the remaining discussion will consider only total hydrocarbon emissions, since these are what most manufacturers have measured and reported.

With the exception of three four-stroke, turbocharged engines (Engines No. 6, No. 16, and No. 4), one gas and the other two dual-fuel, the hydrocarbon emissions do not change much due to the application of $NO_x$ controls. Thus, among the diesel engines, HC emissions generally change by less than 0.1 g/hp-hr. These changes can be either increases or decreases from the starting levels of 0.15 to 0.5 g/hp-hr prior to the application of $NO_x$ controls (Figure 4-52(a)). The two-stroke dual-fuel unit (Engine No. 13) with low emissions of HC experienced at most a 1 g/hp-hr increase in HC emissions from a baseline level of about 1.5 g/hp-hr. Again the application of some $NO_x$ controls did not cause the HC emissions to increase (Figure 4-52(b)). Control of $NO_x$ has a similar effect on HC emissions from gas engines (Figure 4-52(c)) as it does on these emissions from dual-fuel units. As will be mentioned in the next section (4.5), methods exist for reducing

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[8/] Conversion from g/mile to g/hp-hr based on automobile fuel consumption rates of 15 miles/gallon on the road and 0.6 lb/hp-hr on an engine dynamometer.

hydrocarbons from large-bore engines. Therefore, it should be possible to reduce the HC emissions from the three high emission four-stroke, turbocharged engines.

Similar graphs are shown for CO variations due to the application of $NO_x$ controls in Figure 4-53. These graphs show that the incremental changes in the CO emissions generally are less than 2 g/hp-hr. Thus, for the diesel engines shown in Figure 5-27(a), CO emissions increase by less than 0.5 g/hp-hr when the uncontrolled level is low (i.e., around 1 g/hp-hr) and by 1 to 2 g/hp-hr when the uncontrolled levels are higher (3 to 7 g/hp-hr). One exception to this trend is engine No. 60, which is turbocharged. Reductions in its air-to-fuel ratio caused a large increase in CO emissions.

Similarly, dual-fuel engines that are relatively low emitters of CO experience little changes in these emissions due to the application of $NO_x$ controls whereas those units that start from a higher level show greater changes. Specifically, the three engines whose controlled level is around 1.5 g/hp-hr show changes that range from no increases to increases of less than 0.5 g/hp-hr. In contrast, the two four-stroke units whose uncontrolled emissions are 3.6 to 4.5 g/hp-hr react to the application of $NO_x$ controls with CO changes that range from a decrease of about 2 g/hp-hr to an increase of 3 g/hp-hr. With the exception of Engine No. 21 (naturally aspirated), all the gas engines shown in Figure 4-53 experience CO increases less than 1 g/hp-hr when controlled to reduce $NO_x$.

Since the baseline CO emissions from large-bore engines are low, at least by comparison with automobiles, these incremental changes can represent large percentage increases, frequently as high as 100 percent. The highest CO values recorded probably can be reduced by increases in the air-to-fuel ratio along with the control techniques used (usually retard and/or increased

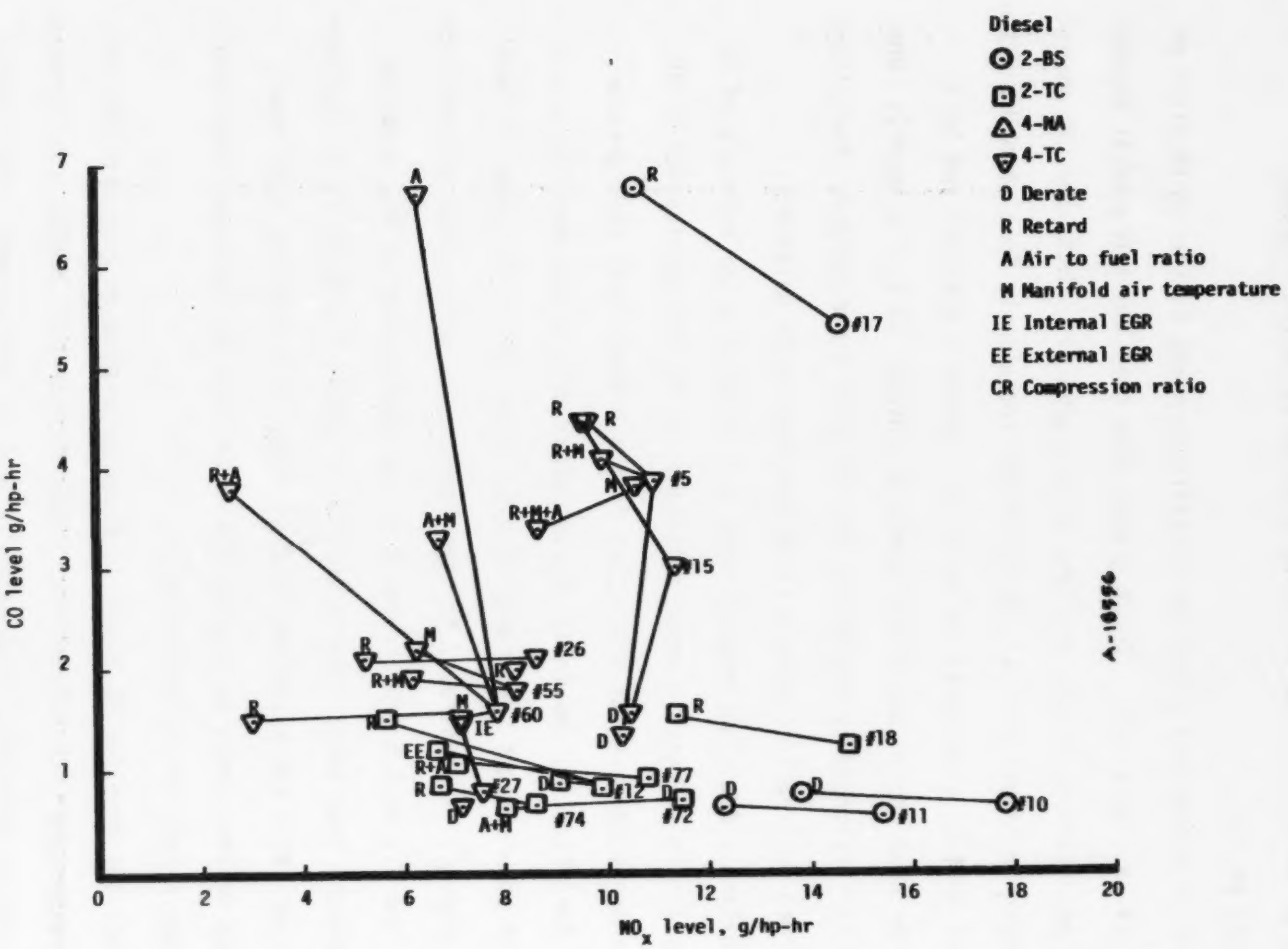

Figure 4-53(a). CO emissions versus controlled $NO_x$ level for diesel engines.

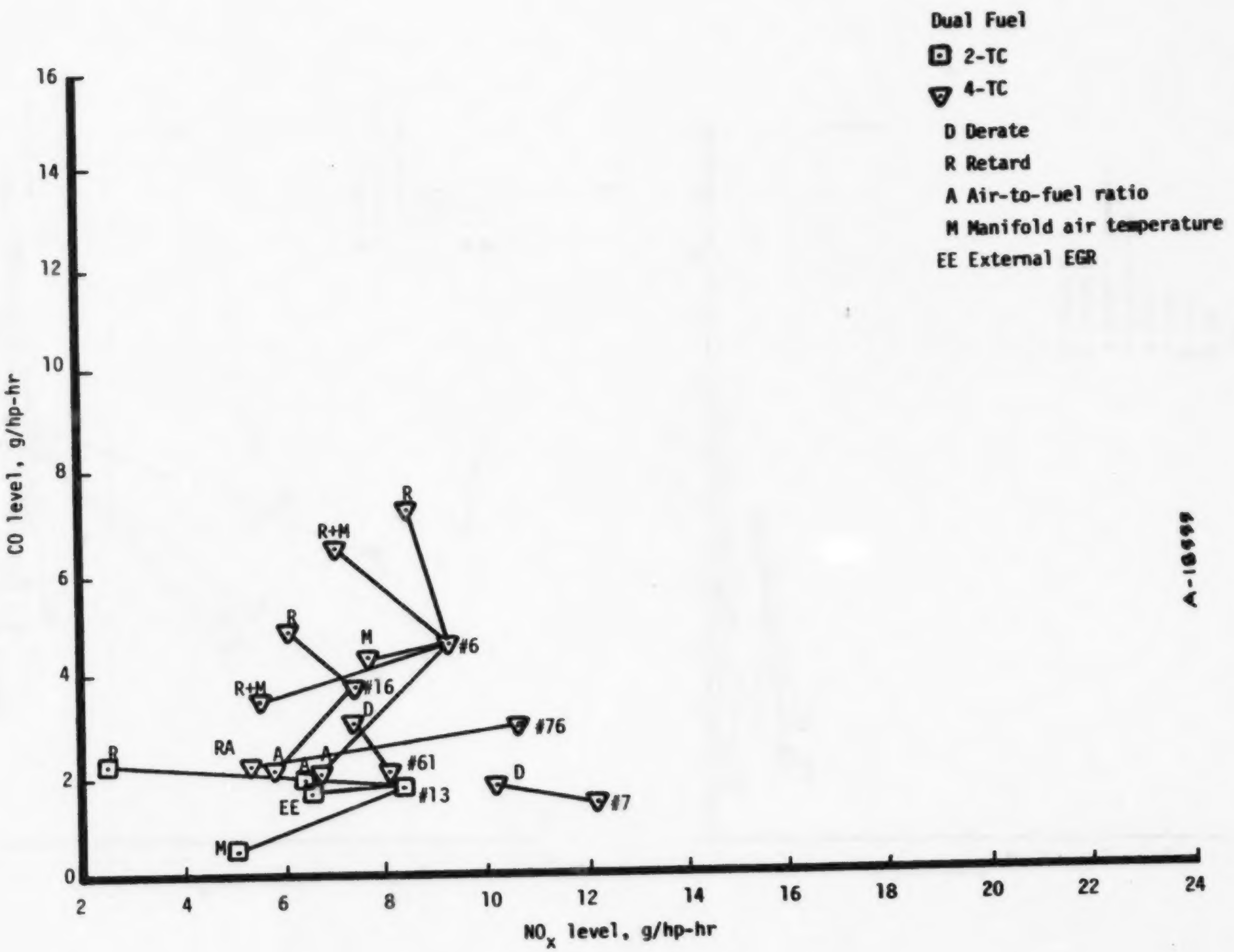

Figure 4-53(b). CO emissions versus controlled NO$_x$ level for dual-fuel engines.

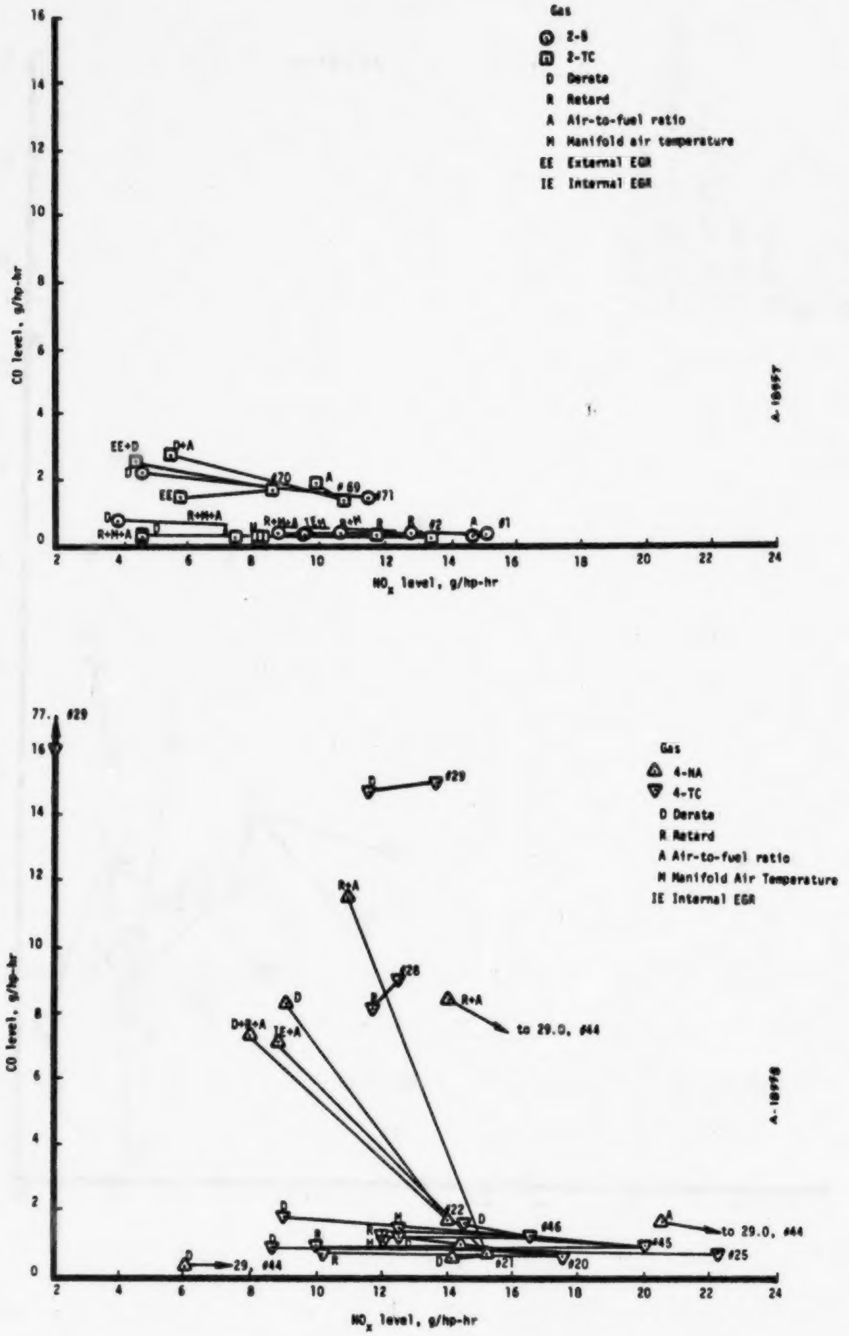

Figure 4-53(c). CO emissions versus controlled NO$_x$ level for gas engines.

inlet air cooling). With this in mind, and if one excludes the two-stroke, blower-scavenged diesel (Engine No. 17), the four-stroke turbocharged diesel (Engine No. 60), the four-stroke turbocharged dual fuel (Engine No. 16), and the four-stroke naturally aspirated gas engine (Engine No. 21) that achieve neither low $NO_x$ nor low CO levels (excluding the dual fuel unit), all the data points for engines with $NO_x$ controls fall below 4 g/hp-hr.

Presumably CO emissions would be reduced from the four-stroke, turbocharged engine (all fuels) as a consequence of any attempt to reduce HC from them. However, if one excludes the data points for Engine Nos. 5, 6, 15, 16, 28 and 60, all other turbocharged engines could meet a 2 g/hp-hr limit on CO, even when controlled for $NO_x$. These figures are significantly lower than the federal standards of 40 g/hp-hr on heavy duty vehicles(140) or even than the proposed 1977 California standards of 25 g/hp-hr(141).

The plumes from most well maintained large-bore engines are virtually invisible when the engine is operating at a steady state. However, excessive retard or large EGR rates will cause diesel and dual-fuel engines to emit smoke. Visible emissions usually were not measured during experiments with $NO_x$ control systems. In most cases the manufacturers observed the exhaust visually and simply did not measure emission rates after the plumes become significantly visible. If the manufacturer did report smoke readings, along with $NO_x$ reduction due to the application of potential controls, the $NO_x$ control results were used in this document only if the plume did not exceed 10-percent visibility.

Figure 4-54 shows the relationships between smoke emissions and $NO_x$ reductions for those engines where data were reported on both pollutants. All of the engines were diesels with the exception of one dual-fueled unit (Engine No. 16). The smoke levels ranged from 1.5 to 8 percent opacity or

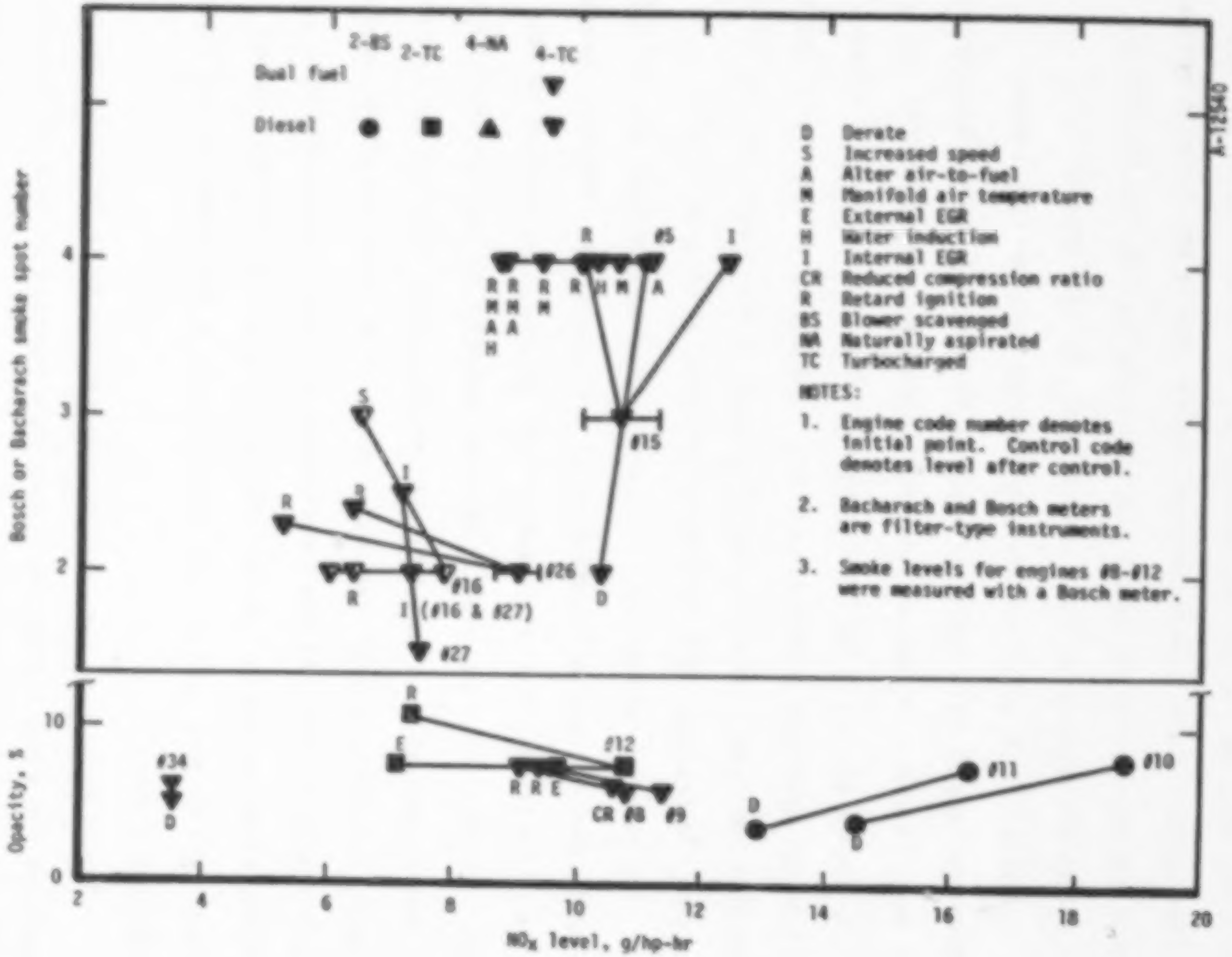

Figure 4-54. Smoke levels versus NO$_x$ levels for large-bore diesel engines.

Bosch Smoke Spot Number 2 to 3 for uncontrolled engines.[9/] Since $NO_x$ controls which caused smoke levels to exceed 10 percent opacity were not considered as acceptable, none of the data points for controlled engines are above this value. However, the effect of progressive application of retard or EGR on smoke is demonstrated best by data which include higher smoke levels. Such data are presented in Table 4-16(142) for two different diesel engines (both two-stroke) and substantiate the above assertion that smoke increases as either the percentage of exhaust gas recirculation or the degree of retard is increased. The table also shows the effect of engine (i.e., stack) size on opacity.

### 4.4.13 Combustion of Nonstandard Fuels

Although most stationary IC engines burn No. 2 distillate or pipeline quality natural gas, both of which contain less than 0.3 percent sulfur (by weight), engines are occassionally sold for applications which use residual oils or unprocessed natural gas. The primary air pollution concern with the use of these fuels is their potential for causing high $SO_x$ emissions, since these fuels typically contain more sulfur then do the distillates and processed natural gas. This section briefly reviews those applications that use nonstandard fuels and their emissions.

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[9/] Since the Bosch Smoke Spot Number is a measure of the soiling characteristics of the particulate and is measured by the darkness of the spot of particulate collected on a piece of filter paper, its value in any given test depends upon such factors as stickiness, color, and reflectence of the particulate. Opacity, on the other hand depends on the size distribution of the particulate, which affects the plume's ability to interfer with the passage of light. Therefore, there is no general relationship between the two systems.

TABLE 4-16. RELATIONSHIP BETWEEN SMOKE, EGR, RETARD, AND ENGINE SIZE

| Engine Type[a] | Control[b] | Opacity, % | |
|---|---|---|---|
| 2-BS-D | None | 4.7 | |
| | 10% EGR | 12 | |
| | 20% EGR | 27.5 | |
| | 30% EGR | 59 | |
| | 4° advance[c] | 2.7 | |
| | None | 4.6 | |
| | 4° retard | 10 | |
| | | 6 Cylinder | 12 Cylinder[d] |
| 2-TC-D | None | 7.5 | 11.0 |
| | 4.9% | 10.0 | 14.0 |
| | 8.4% | 11.5 | 16.7 |
| | 12.1% | 14.8 | 21.4 |

[a] Blower-scavenged engine is two-cylinder test engine with needle valve injector (from Reference 142). Turbocharged unit is Engine No. 12.

[b] All EGR data based on hot EGR.

[c] Injection advance is not a control; data included to show trend.

[d] Computed from six-cylinder data using Beer Lambert law (see Figure 8-1), double the stack cross sectional area (to accommodate doubling of exhaust volume), and the same particulate concentration ($mg/m^3$) as in six-cylinder unit.

Some of the more common applications of engines using unprocessed natural gas are sewage treatment plants. The engines at these facilities often burn sewage digestor gas. Although the sulfur content of this gas varies, typical values range from 100 to 150 ppm of hydrogen sulfide ($H_2S$)(143). This level translates into an emission rate of less than 0.5 g/hp-hr of $SO_2$ (assuming 600 Btu/$ft^3$ gas and a fuel rate of 7500 Btu/hp-hr). Emissions data shown in Table 4-17(144) for a sewage plant in Los Angeles, however, indicate that some engines emit as much as 1 g/hp-hr of $SO_2$(145). These levels are well below the estimated 2 g/hp-hr emitted from a diesel engine burning 0.5 percent sulfur distillate (see Section 4.1.6).[10/]

Stationary reciprocating engines occasionally are sold with the capability of burning residual or crude oils. The only known domestic site where stationary IC engines are burning residual oil is the Anamax Mining Company diesel electrical generation plant near Tucson, Arizona(146). This installation was designed to take advantage of the lower cost of these heavy oils as well as their availability at that site. Distillation of crude oil produces distillates which are relatively low in sulfur content and, hence, concentrates any sulfur present in the crude into the residual oil. The sulfur content of crude and residual oils depends on their source and, in the case of residual oil, on how severely the parent crude is distilled. Table 4-18(147) indicates sulfur contents of crude oils from various sources.

The naturally low sulfur, or "sweet", crudes and their residuals may have sufficiently low sulfur content to present no $SO_2$ emission problem, but are in short supply and command higher prices than the high sulfur fuels. The

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[10/] The No. 2 distillate available to domestic markets has a sulfur content which averages 0.25 percent (by weight) and ranges from 0.01 to 0.75(140).

TABLE 4-17. EMISSIONS FROM ENGINES BURNING SEWAGE DIGESTOR GAS
(from Reference 144)

| Engine # | HP | # cyl | rpm | Fuel | | Emissions, g/hp-hr | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | % $CH_4$ | Heating Value | $NO_x$ | CO | $HC_T$ | $SO_2$ |
| 1 | 1100 | 10 | 330 | 62 | 580 | 3.14 | <0.1 | 0.96 | 0.10 |
| 2 | 1100 | 10 | 330 | 60 | 557 | 8.43 | <0.1 | 0.91 | 0.12 |
| 3[a] | 1600 | 8 | 352 | 64 | — | 2.21 | <0.1 | 9.16 | 1.08 |
| 4[a] | 1600 | 8 | 350 | 64 | — | 2.37 | <0.1 | 5.65 | 1.03 |

[a] Dual-fuel operation, 4.5 gallon/hr No. 2 diesel; 14,600 scf/hr gas.

TABLE 4-18. SELECTED APPROXIMATE PROPERTIES OF CRUDE OILS CURRENTLY GENERATING U.S. RESIDUAL OIL SUPPLY (Reference 147)

| Origin | Property | | | | | |
|---|---|---|---|---|---|---|
| | Vanadium (ppm) | Nickel (ppm) | Sulfur (wt %) | Nitrogen (wt %) | Residual Percent (@ 787°F) | Residual Carbon (wt %) |
| Venezuela | 100-900 | 30-100 | 1-6 | 0.2-0.6 | 30-60 | 10-15 |
| Libya | 7 | 10 | 0.3 | 0.15 | 25 | 8 |
| Mid-East | 20-50 | 5-15 | 1-3 | 0.1 | 30 | 10 |
| Indonesia | — | 20 | 0.1 | 0.1 | 50 | 5 |
| Alaska (North Slope and adjacent) | 15-30 | 5-15 | 1 | 0.2 | 35 | 10-15 |
| California | 100-200 | 100 | 1-5 | 0.6-0.9 | 60 | 10 |

sulfur content of residual oil can be reduced by catalytic hydrodesulfurization, a petroleum refining technique. Costs for this refining are discussed in Section 8.2 for different amounts of fuel sulfur.

Residual oils, blends, some crudes, and heavy distillate also contain higher amounts of fuel-bound nitrogen than the lighter distillates. Therefore, the combustion of these fuels may result in greater $NO_x$ emissions than if lighter distillates were burned. However, a byproduct of catalytic hydrodesulfurization is a reduction in both the metal and nitrogen content, as well as the sulfur, of the residual oils. Thus, fuel oil desulfurization processes can reduce fuel nitrogen. In one case, using a high sulfur, high nitrogen residual fuel feedstock, 78 and 94 percent sulfur removals were accompanied with 40- to 70-percent reductions in nitrogen, respectively(148). At this time, however, there are no known emissions data for stationary IC engines burning heavy fuels. Thus, the effect of heavy fuel combustion on $NO_x$ and other emissions is unknown.

## 4.5 EMISSION REDUCTION TECHNIQUES PRIMARILY FOR HC AND CO

As mentioned in Subsection 4.4.12 hydrocarbon (HC) and carbon monoxide (CO) emissions from large-bore engines are generally low, especially when the engines are operated at steady state. Therefore, very little effort has been expended by manufacturers to reduce the emission rate of these pollutants, except where ancillary to a smoke reduction program. One foreign manufacturer of medium-bore, high-speed engines has developed a relatively low HC and CO engine for use in mines (0.4 and 1.8 g/hp-hr, respectively, based on the 13 mode composite test cycle)(149), and at least one domestic manufacturer of catalysts markets a HC and CO catalytic oxidation unit for truck sized diesel and gas-fueled engines(150). When installed on a diesel, these catalysts bring about reductions of 70 percent in CO emissions and 45 percent in HC emissions while fuel consumption, peak smoke, and $NO_x$ emissions remain essentially unchanged(151). Furthermore, catalytic systems to reduce HC and CO emissions from heavy duty gasoline and diesel have remained 80 percent to 95 percent effective for both gasoline and diesel engines throughout the 1000-hour test(152,153). No attempt to extend either of these approaches to large-bore engines has been reported. Therefore, the description of the HC and CO control techniques which are presented below are mostly qualitative and frequently draw upon experience gained with medium-bore, medium-to-high speed engines.

In general, the combustion process is enhanced by high temperatures, freely available oxygen, good mixing of the fuel-air mixture, and long residence times. Since these conditions also result in the highest $NO_x$ formation rates, care must be exercised in a balanced emission reduction effort not to reduce HC and CO at the expense of $NO_x$. Therefore, attempts to reduce HC and CO emissions must concentrate on quench layer effects, air-to-fuel mixture inhomogeneities, fuel droplet size, exhaust gas treatment, and the like.

The three major approaches to control of HC, (1) modifying air and fuel handling systems, (2) modifying combustion chamber designs, and (3) treating the exhaust, are discussed in detail in the following subsections.

### 4.5.1 Modifications to Air and Fuel Handling Systems

This category of HC and CO emission control techniques can be further divided into the three subcategories listed below:

1. Air-to-fuel mixture
2. Liquid fuel injection (compression ignition engines only)
3. Cylinder scavenging

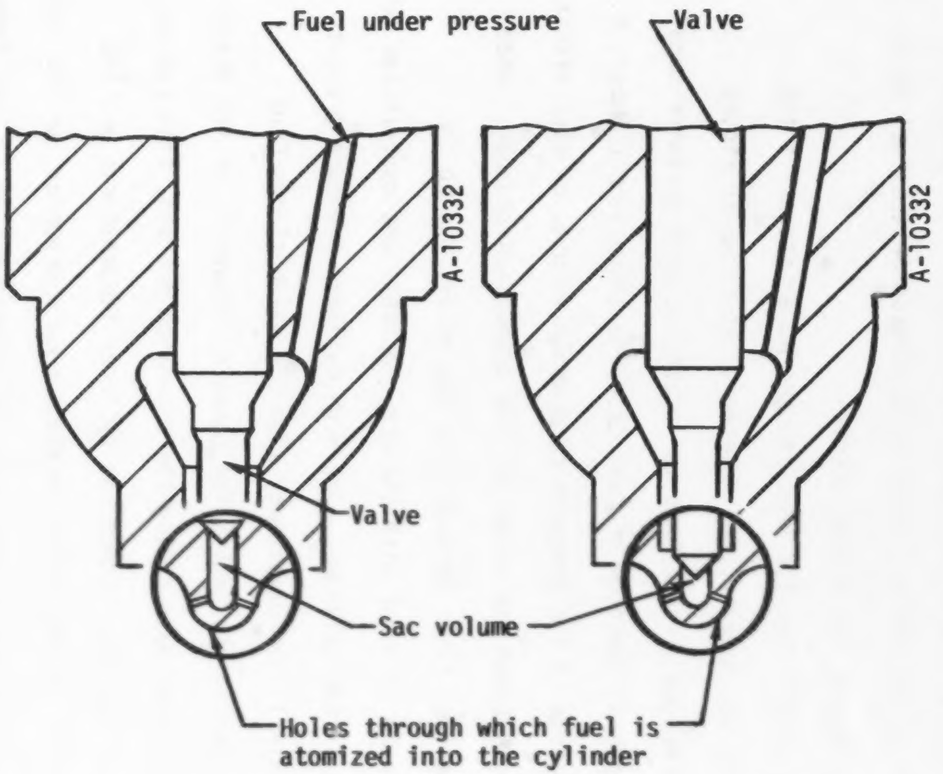
Since HC and CO emissions decrease as the combustible mixture becomes leaner, at least up to a point (see Figure 4-28), increasing the air-to-fuel ratio can be a control technique for these pollutants. On a diesel engine this usually requires the addition of a turbocharger, if the engine is not already equipped with one, or an increase in the capacity of the existing turbocharger unit. In a carbureted spark-ignition engine, the air-to-fuel ratio can be changed by alterations to the venturis and fuel nozzles in the carburetor. Unfortunately, as shown in Figure 4-28, both minimum HC and maximum $NO_x$ emissions frequently occur at about the same air-to-fuel ratio. Hence this approach has limited utility in a total emission reduction program, unless coupled with other controls.

Some carbureted gas engines, however, are adjusted to operate near stoichiometric, on the average, to overcome the problem of poor distribution of fuel and air among the cylinders(154). These same engines would have excessively lean mixtures in some cylinders if the carburetor were not set rich, and these lean cylinders could not support combustion (i.e., they would exceed the lean misfire limit). This problem can be ameliorated by improved aerodynamic design of the carburetor and intake manifold system or by converting the engine to a fuel injection system. In fact, most large-bore, gas fueled engines already use fuel injectors.

The injector in a liquid fueled compression ignition engine is required to deliver fuel only when specified, to atomize the fuel into droplets of the smallest possible diameter, to distribute these droplets uniformly throughout the combustion volume, and to avoid impinging fuel on the relatively cool cylinder walls (potential quench layer). Therefore, attempts to reduce hydrocarbon (and smoke) emissions have traditionally concentrated on higher injection pressures to improve atomization and on changes to the design of the injector tip, in particular, hole size, location, and number(155). Frequently these efforts have gone hand-in-hand with changes to the shape of the piston head, because wall impingement problems can sometimes be reduced by directing a portion of the fuel spray onto the contoured surface of the piston head.

A major change to the injector that reduces HC emissions is the introduction of the low sac nozzle (LSN) injector. As shown in Figure 4-55(156), older nozzles use a valve that is located at some distance from the injector tip to control the flow of fuel into the combustion chamber. A consequence of this design is the presence of a small volume, or sac, within the injector that lies between the valve and the combustion chamber. When the valve is open, the fuel flows through this sac on its way to the combustion chamber, where it mixes with the air, ignites, and contributes to a rapid pressure rise in the cylinder. The high cylinder pressure prevents any fuel that is in this little passageway when the valve closes from entering the cylinder during the combustion process. However, as the piston moves away from the cylinder head during the expansion process, this remaining fluid "dribbles" out of the sac. Since the cylinder temperatures are too low to support combustion by now, these droplets remain unburned, or are only partially oxidized, and eventually pass into the exhaust system in that form.

Several manufacturers have practically overcome this problem by switching to a nozzle with a greatly reduced sac volume, i.e., a low sac nozzle(157). A schematic representation of such an injector is shown in Figure 4-55. In some designs of sac volume is so small as to be virtually negligible.



a. Old standard production spray tip.

b. Low sac volume spray tip.

Figure 4-55. Schematic illustrations of standard production spray tip and the low sac volume spray tip. (after Reference 157).

The use of such nozzles lead to 30- to 35-percent reductions in HC and CO(158). The reductions were accompanied by a decrease in opacity to 4 percent (at rated conditions); $NO_x$ emissions, however, rose nearly 35 percent, probably as a result of changes in the injector tip design that accompanied the switch to a low sac volume. These changes could alter the fuel spray pattern and/or degree of atomization.

Unburned or partially burned hydrocarbons that come from either injector "dribble" or quench layers are usually concentrated in the last volumes of gas to be expelled from the cylinder during the exhaust stroke. Therefore, EGR, whether internal or external, serves to reduce HC emissions, in addition to $NO_x$, as long as the recirculation rate is not so great that portions of the combustion reaction are quenched. Reductions in the scavenging pressure in a two-stroke, blower-scavenged engine can also have the same effect.

### 4.5.2 Combustion Chamber Modifications

When one discusses HC and CO control techniques for large-bore engines, one must remember that these contaminants are present only at very low quantities (typically below 2.0 g/hp-hr) in the exhaust from these units. This is so mainly because of the presence of excess air during combustion. In addition, the high temperatures and relatively long residence time (low engine speed) facilitate the combustion of the fuel. Finally, the large volume, low surface-to-volume ratio, and use of injectors gives the engineer of medium-and-large-bore units more control on the following parameters than he has with automobile-sized engines: air-to-fuel distribution within a cylinder, variation in air-to-fuel ratio among cylinders, and fuel droplet impingement on the cylinder walls. As a result, an average value for the total HC emissions from a variety of large engines is 2 g/hp-hr (see Figure 4-52) which corresponds to 0.1 percent of the incoming fuel.[11] Thus, even though combustion characteristics are evidently very good from the point of view of HC and CO, combustion chamber design can be important as one attempts to oxidize the last 0.1 percent of the fuel.

A variety of chamber designs has already been presented in Section 4.4.8. Of these techniques, higher swirl, inlet valve and/or port redesigns, and contoured pistonheads reduce HC and CO because they improve the mixture rate and homogeneity of the fuel and air. The designs for staged combustion, such as precombustion chambers and "squish lips", can reduce HC and CO, too. These geometries cause the combusting mixture to expand the available oxygen with the unburned or partially burned fuel. Unfortunately this mixing process is also the source of some increased fuel consumption; as the burning mixture expands into the main chamber, its pressure is decreased slightly, resulting in reduced power output. The mechanical energy lost by the gas becomes thermal energy and is seen as a slight increase in the gas temperature. Of the 5- to 8-percent energy loss attributed to precombustion chambers (see Section 4.3.8), however, most is probably due to the heat loss in the precombustion chamber.

Another technique that is available to decrease HC emissions, especially from diesel engines without affecting $NO_x$ emissions, is to reduce the "quench layer", or region immediately adjacent to the combustion chamber surfaces.

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[11] A comon assumption used in calculating mass emissions of total hydrocarbons is that a typical hydrocarbon molecule emitted in the exhaust is the same as a typical fuel molecule(159).

Heat transfer outward through the cylinder walls causes the mixture temperatures in this layer to be too low to support complete combustion.(160) Some unburned hydrocarbon emissions in diesel engines result from fuel droplets that were injected into this "quench layer". Attempts to reduce this source of HC emissions rely primarily on injector design to avoid impinging the fuel droplets onto the cylinder walls and on relocation of the uppermost piston ring as close to the piston head as possible to minimize the relatively cool space between the piston and the cylinder adjoining the combustion volume(161).

These quench layer hydrocarbons could theoretically be reduced by using higher cooling water temperatures. However, the increase in wall temperatures may give rise to structural problems (current cooling systems are presumably designed to avoid such problems) and, in any case, the resulting higher combustion temperatures would result in an increase in $NO_x$ emissions.

Since high temperatures are conducive to complete combustion and, hence, low HC and CO emissions, these pollutants can be minimized by raising the peak combustion temperature through an increase in the compression ratio. This parameter is the ratio of the largest volume in the cylinder to the smallest -- i.e., the ratio of the volume when the piston is at the bottom of its stroke (bottom dead center or BDC) to that when it is at the top (TDC). Increasing this ratio increases the volume change and hence the temperature rise of the fuel and air during the compression stroke. This approach also increases the efficiency of the engine, but unfortunately at the expense of higher $NO_x$ emissions.

### 4.5.3 Exhaust Treatment

Potential methods of treating the exhaust to reduce $NO_x$ emissions have been discussed in Section 4.4.9. Here we are concerned only with catalytic converters, thermal reactors, and exhaust manifold air injectors, which are designed to reduce HC and CO emissions. Although these systems have been tested on small- and medium-sized engines, to our knowledge they have not been used on large-bore units. Application of these techniques to large engines require careful design to insure that proper operating temperatures are maintained in all parts of the larger units. The oxidation reactions which convert the HC and CO to water and $CO_2$ take place only above a certain temperature, and this high-temperature environment is maintained inside the exhaust treatment unit by the heat from earlier reactions. It is more difficult to insure the existence of a homogeneous high-temperature environment throughout the combustion volume of a large unit than of a small one because the heat release from localized reactions is absorbed by more gas in the large system than in the small one. That is, the heat generated may not affect the entire volume in the large unit. Nevertheless, there does not seem to be any technical reason why such units cannot be designed.

The catalytic converter has been proven on mobile gasoline engines and functions by causing the oxidation reaction to continue at reduced temperatures. It is a passive device which consists of a container that houses a porous structure covered with the catalyst material. The converter replaces a portion of the exhaust tail pipe. The chemical reaction then takes place on the catalyst-coated surface as the exhaust passes through the converter. If automotive type platinum catalytic converters are used, only unleaded fuels can be burned to avoid inactivating the catalyst. In addition, fuels with low sulfur content should be used to suppress sulfuric acid formation ($H_2SO_4$). This technique is not limited to small-bore engines and could, therefore, be applied to large-bore units as well. The need to design the converter to operate at uniform elevated temperature has already been discussed; cost implications are considered in detail in Section 8.2.

In the thermal reactor air is added to the exhaust stream in a container which is located in the exhaust pipe system and is specially designed to maximize both the residence time of the charge and its turbulence. The purpose of the device is to provide a site where oxidation can proceed at elevated temperatures. These are maintained by the heat released from the oxidation reaction of CO and HC with excess air. Air injection into the exhaust manifold functions to the same end as the thermal reactor. However, it is not as effective an oxidizer of HC and CO as the thermal reactor because it utilizes the existing shape of the exhaust system. This volume is generally not optimized for maximum residence time, heat retention, or turbulent mixing.

## 4.6 SMOKE

In general smoke in well operated and maintained large-bore engines is a problem only for compression ignited engines when fired with nothing but liquid fuel. Attempts to reduce visible emissions aim at reducing locally rich zones by improving air-to-fuel mixing and distribution throughout the cylinder. The following techniques are most commonly used for this purpose:

- Improved combustion through chamber redesign, particularly piston head shape and increased swirl
- Leaner air-to-fuel ratio (by carburetor adjustments, inlet manifold design, or turbocharging)

- Improved fuel atomization and dispersion
- Low-sac volume injectors

A number of manufacturers have successfully reduced smoke emissions over the operating range of large diesel engines through the use of improved needle and low-sac fuel injector designs. Visible emissions from most current, properly maintained units are well below 10-percent opacity during continuous load application(162,163).

Figures 4-56(164) and 4-57(165) illustrate the improvements obtained by the Electromotive Division (EMD) of General Motors on both blower-scavenged and turbocharged versions of their two-stroke Model 645 diesel engine (although this engine is used mainly in locomotives, it is also sold for stationary applications). Smoke reductions were also achieved on these engines by widening the size of the ports in the cylinder liners 12-1/2 percent to improve charging and scavenging processes and by moving the top piston ring ("fire ring") three-quarters of an inch higher. Sea level smoke emissions at rated conditions are now about 6-percent opacity for both engines, having been reduced from 20 percent for the turbocharged unit and 30 percent for the blower scavenged one. The effect of derating is shown in Figure 4-56 to illustrate that this technique has to be used with blower-scavenged engines at high altitude in order to comply with local regulations that restrict visible emissions to a Ringelmann No. 1 (20-percent opacity).

Although not recommended by manufacturers, additives are sometimes mixed with the fuel to reduce smoke. Barium is a common additive and

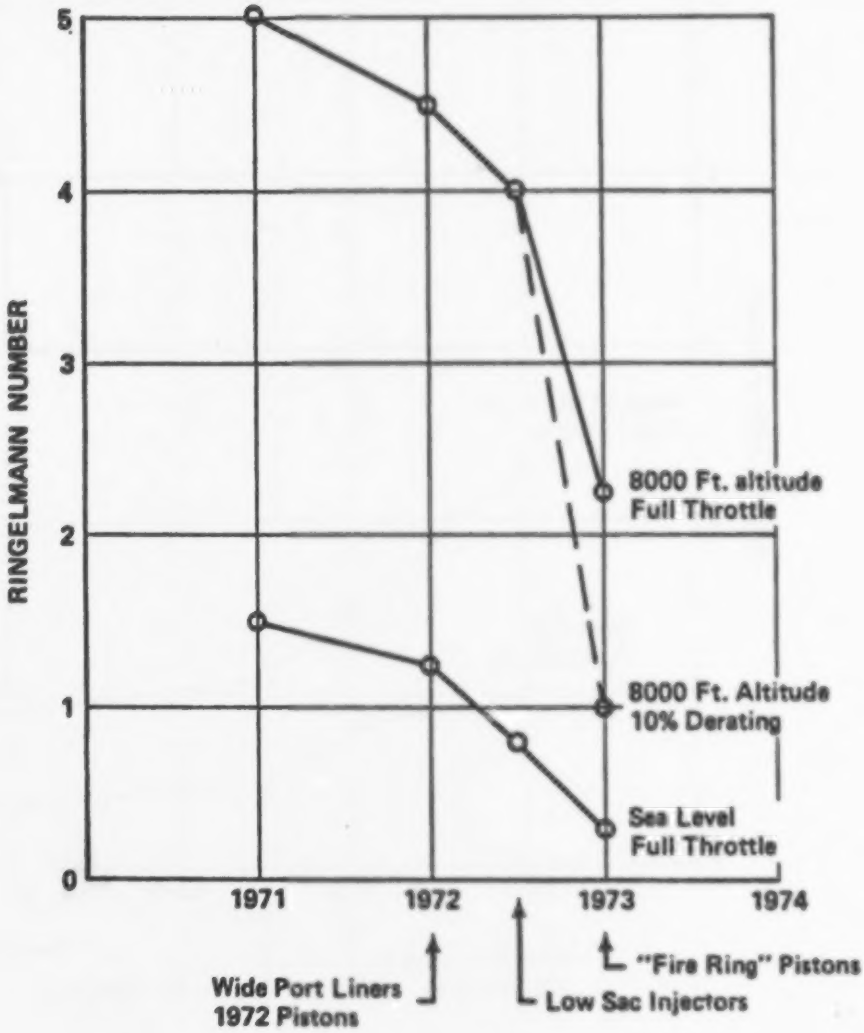

Figure 4-56. Exhaust smoke reduction schedule for EMD Model 645 Roots blown locomotive engines (164) at rated conditions (Ringelmann 1 = 20% opacity, No. 2 = 40%, etc.).

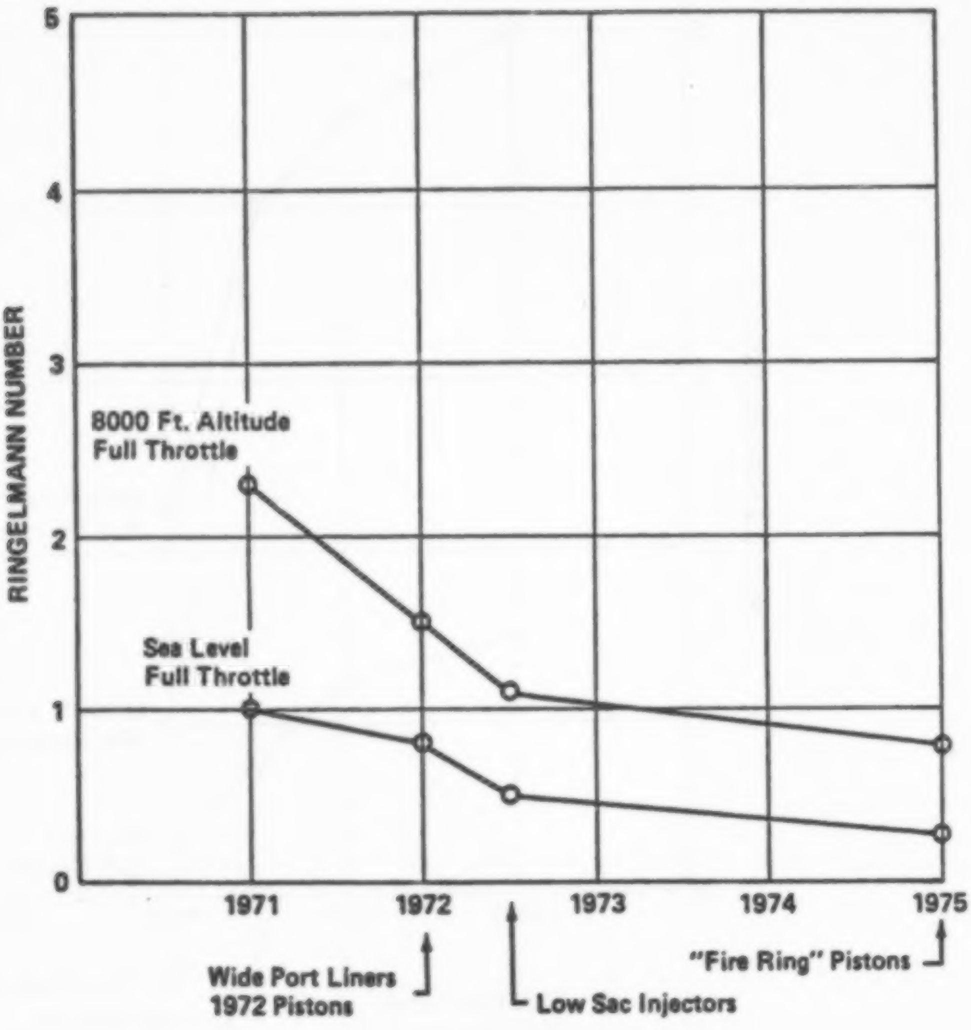

Figure 4-57. Exhaust smoke reduction schedule for EMD Model 645 turbocharged locomotive engines (165) (Ringelmann 1 = 20%, opacity, No. 2 = 40%, etc.).

reportedly reduces the particulate mass emission rate[12/] in medium-bore diesel engines by as much as 50 percent(166-168). This is theorized to be due to the inhibiting effect of the barium on the formation of carbon in the vicinity of the burning fuel droplets, or to its action as a catalyst accelerating the oxidation of the carbon formed(169).

In addition to reportedly decreasing the mass of particulate emitted, the character of the particulate composition is altered as well by the additives(170). This was observed in one case where carbon comprised 96 percent, hydrogen 2 percent, iron 1 percent, and zinc 1 percent of the particulate sampled when untreated fuel was used. In contrast, carbon comprised only 36 percent and hydrogen 1 percent of the particulate mass emitted when a barium treated fuel was substituted. Barium sulfate ($BaSO_4$) now represented about 60 percent of this mass, and iron and zinc were present in trace quantities only.

Despite the effectiveness of this approach, it is not favored by industry because of concern over possible build-up problems that might occur at the cylinder ports as a result of the increased metallic content of the fuel(171,172). Furthermore, $BaSO_4$ and the other soluble barium compounds produced may represent a health hazard when emitted into the atmosphere in substantial quantities.

Although no particulate scrubber is known to have been used on large-bore engines, probably because they usually operate at steady state and generally emit little smoke, one automobile-oriented service industry has developed such a device to remove lead particulates from automobile

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[12/] The level of smoke opacity was determined using a Hartridge Smokemeter. The weight of solid particulates was measured by filtering a precise volume of exhaust gas through glass filter papers and weighing the deposited soot.

exhaust(173). In tests on four cars equipped with these particulate traps the average air-suspended lead emissions were reduced about 67 percent from 26.6 to 10.1 percent of the lead contained in the fuel. The same particulate trap also reduced total particulate emissions (one-fifth of which is lead) by 74 percent.

Presumably, a similar system, or one of the many kinds of particulate scrubbers available, could be used if it were decided to reduce $NO_x$ significantly by means of large amounts of retard or EGR. Both these techniques reduce $NO_x$ effectively, but, beyond a certain point, only at the expense of smoke.

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# CHAPTER 5

# MODIFICATIONS AND RECONSTRUCTIONS

## 5.1 BACKGROUND

This chapter identifies and discusses possible or typical changes to stationary reciprocating internal combustion engines which could be termed modifications or reconstructions. According to Federal Regulations, such changes would subject an engine to a standard of performance. A modification is defined as "...any physical change in, or change in the method of operation of an existing facility which increases the amount of any air pollutant (to which a standard applies) emitted into the atmosphere by that facility or which results in the emission of any air pollutant (to which a standard applies) into the atmosphere not previously emitted"(1). An "existing facility" is defined as one which would be required to conform to a standard of performance, if it were new, but which was, in fact, constructed or modified before the date of proposal of the standard of performance.

The regulation requires the owner or operator of any engine classified as an "existing facility" to notify the EPA of changes which could cause an increase in emissions of an air pollutant for which a standard of performance applies(2). These changes would not be termed "modifications" -- i.e., the existing facility would not have to meet the standards of performance -- if the owner or operator could demonstrate that no increase in emissions for which a standard applied resulted from the alteration.

In addition, the regulation also defines the term "reconstruction," namely the "...replacement of a substantial majority of the existing facility's components irrespective of any change of emission rate"(3). Therefore, if substantial numbers of parts are changed in an engine which belongs to the class of existing facilities, this engine will be considered to have undergone a reconstruction and will have to conform to standards of performance as if it were a new engine. The purpose of this provision is to discourage the perpetuation of a facility which, in the absence of a standard, would normally have been replaced(4). Again, the regulation requires the owner or operator to provide information concerning the construction or reconstruction of an existing facility to EPA(5).

Certain practices, however, are exempted from being considered as a modification(6). These exemptions are listed below, in terms that are appropriate for stationary engines.

1. Routine maintenance, repair, and replacement of parts
2. Increases in power generation, provided such increases do not exceed the rated power of the engine
3. Increases in hours of operation
4. Use of an alternative fuel, if specified by the manufacturer for that particular engine
5. Addition of a control device which reduces emissions or replacement of a control device by another of at least equal efficiency
6. Relocation or a change in ownership of an engine

The addition of a new, modified, or reconstructed unit to a multiple-unit installation subjects only the addition to a standard of performance and not the remainder of the existing facility(7).

Specific changes to stationary, reciprocating internal combustion engines that could be classified as modifications or reconstructions will be discussed separately in the following two sections.

## 5.2 MODIFICATIONS

### 5.2.1 General

As discussed in Section 5.1, the law regarding modifications to stationary reciprocating engines is very explicit; that is, any physical or operational changes to an engine that result in increased emissions (for which a standard applies) subjects that engine to a standard of performance. This section will discuss those physical and operational changes to stationary engines that could result in increased emissions. Alterations which are likely candidates to be exempted as modifications are noted. Physical changes are alterations to either the engine's structure, its components, or its auxiliaries, and operational changes include variations of such parameters as fuel, air-to-fuel ratio, ignition or fuel injection timing, and manifold inlet air temperature.

### 5.2.2 Physical Changes

In general, stationary engines are "customized" for a particular application, and major changes are not made to their hardware and components during their lifetime other than routine replacement of wearing parts(8-11). Practically, then, modifications will usually occur only when a replacement part has been altered from the original design, and the use of this part causes the engine to produce greater emissions than the original configuration. Manufacturers have, in fact, made changes in recent years, primarily to improve fuel economy and smoke characteristics of their engines(12,13). For example, one manufacturer of medium-bore diesel-fueled engines has made four injector design changes since 1962(14). Although many of these changes may reduce HC, CO, or smoke levels, they frequently will increase $NO_x$ levels. Hence, if a user replaces an old part with a newer, modified one, he may be making a change that is a "modification" as defined by Federal Regulations. Examples of such changes are discussed in the following paragraphs.

The design of various engine components can significantly affect emissions from an engine. For example, the cylinder head shape, the compression ratio, or the shape of the piston may be altered to enhance combustion(15). Since improved combustion frequently results in higher temperatures within the cylinder, these design changes can cause $NO_x$ emissions to increase. Another alteration is to move the uppermost piston ring to a higher location on the piston. This change reduces the quench layer, and hence hydrocarbon formation, but generally does not affect emissions of the other pollutants. One study, however, indicated that moving the uppermost piston ring higher on the piston and increasing the injector rack setting substantially reduced CO in addition to decreasing HC and caused the $NO_x$ level to increase(16).

Engine components used for the intake of fuel and air or the exhaust of combustion products may also influence emissions. The number and size of the holes in a fuel injector as well as the sac volume (see Figure 4-55) have an effect on emissions(17). If the fuel injector change improves the combustion characteristics, $NO_x$ emissions may increase. Decreasing the sac volume, in general, reduces HC emissions without affecting the other pollutants. Fuel pump characteristics and intake valve camshaft design determine injection rates, which in turn affect combustion characteristics and, hence, emission rates (see the discussions on injection timing and internal EGR in Chapter 4).

In spark-ignition engines the carburetor, intake manifold, and intake valves (location and number) affect the mixing of fuel with air (both local and overall air-to-fuel ratio, see Sections 3.2.1 and 4.1 through 4.4). Consequently, changes to these systems may increase the emissions from the engine, particularly $NO_x$. The location, size, and number of ports in the liners of a two-cycle engine influence the amount of air in the cylinder and its manner of mixing with the fuel(18). Similarly, blowers, turbochargers, and aftercoolers determine the delivery rate and temperature of the inlet air or the air and fuel mixture (see Sections 3.2.1 and 4.1 through 4.4).

The number and location of exhaust valves also affect emissions. For example, a manufacturer of two-cycle engines uses heads with four exhaust valves on engines that must meet EPA or California heavy-duty vehicle emission standards, but retains the original two-valve design for the uncontrolled stationary counterparts of these truck engines(19). Variations in valve overlap (four-cycle) or port overlap (two-cycle) can alter the emission characteristics of an engine, too. The amount of overlap affects the amount of oxygen available for combustion as well as the temperature of the air or the air and fuel mixture. Most of these items could be changed by the user on an existing engine and, therefore, might be a modification in the present context.

Large and medium-large engines (>500 hp) that are intended for stationary applications are occasionally sold without much of the auxiliary equipment, such as radiators, turbochargers, aftercoolers, etc. Therefore, if the purchaser does not use a correctly-sized cooling system (radiator or cooling tower), or an undersized aftercooler on a turbocharged unit, the engine could run hotter than specified by the manufacturer. In either case, the temperature of the mixture in the cylinder would be higher than specified by the manufacturer, and hence the $NO_x$ emissions would also be higher. In other words, the use of auxiliary equipment that causes peak cylinder or inlet manifold air temperatures to be higher than specified by the manufacturer could be considered a modification.

Facilities with many engines usually have a staff of engineers and technicians that are skilled in the operation and maintenance of their engines. These users may occasionally change portions of an engine to eliminate problems with the engine or to improve its performance. Presumably these alterations involve only peripheral equipment such as radiators, lubricating oil feed systems, etc., and hence have, at most, a secondary effect on emissions. One user, for example, was able to reduce visible emissions from large, two-stroke natural gas-fueled engines by changing the lubricating oil feed system(20). It is unlikely, however, that these users would redesign such emission sensitive components as the fuel injector, intake valve or port geometry, piston head shape, cylinder head shape, etc.

### 5.2.3 Operating Changes

As discussed in Sections 3.2.2.3 and 4.1 through 4.4, emissions from an engine depend, in part, on its operating conditions. Some of these changes are a consequence of the normal operation of an engine over the range of conditions for which it was designed. For example, variations in torque, speed, output power, or air-to-fuel ratio that are within the engine specifications belong in this category of changes. In contrast, injection or ignition retard or advance from the manufacturer's specification or purposeful alteration of the air-to-fuel ratio do not follow from typical usage patterns and, therefore, could be considered modifications.

The use of fuels not specified by the manufacturer for a particular engine could also constitute a modification. Such a condition would arise, for example, if an owner or operator began to fuel an existing engine, which is specified to burn only No. 2 diesel oil, with No. 6 oil.1/ In general, though, manufacturers specify a range of fuels to be used in their engines. In fact, precombustion chamber engines are specifically designed to accommodate a variety of light fuels and dual-fuel engines are designed to operate in either of two modes: full diesel or dual-fuel operation. Therefore, although fuel type and quality can effect emissions (Section 3.2.2.3.4), the use of alternative fuels would constitute a modification only if the change were to a fuel not originally intended for use in the engine when it was manufactured.

## 5.3 RECONSTRUCTION

A reconstructed engine, as discussed at the beginning of the chapter, is essentially an engine which has undergone a major rebuilding when it would otherwise have been scrapped or replaced with a new engine. It is difficult to apply the definition of reconstruction ("...replacement of a substantial majority of the existing facility's components...") to a stationary engine because substantial portions of large- and medium-bore engines are replaced, in practice, as a matter of routine maintenance. That is, stationary (and industrial mobile) engines are currently given several major overhauls during

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1/At the present time there is only one known domestic installation of engines burning residual fuel.

the normal life of the engine without any stimulus from emission regulations. The problem, then, is to distinguish routine maintenance and overhaul from the rebuilding of an engine to avoid a standard of performance.

Large engines receive several minor and major overhauls during the lifetime of the engine(21-24). Minor overhauls are performed after 8,000 to 16,000 hours and include new rings, new or reground valves, and occasionally new or rebuilt pistons. After 20,000 to 80,000 hours of continuous service these large-bore engines are given a major overhaul.[2] At this time the cylinder liners (which are removable from the engine block) are replaced in addition to the items covered under a minor overhaul. Generally, the cylinder head, the crankshaft, rods, main bearings and rod bearings will last indefinitely; nevertheless, these parts do sometimes fail, usually due to lubrication failures, which can occur when the oil becomes contaminated, or to extensive operation at overload conditions. However, the basic engine frequently remains in service indefinitely.

In reality, several factors discourage perpetuation of these engines. First, it can become difficult to obtain manufacturer service and replacement parts for older engines. Second, power requirements change with time and frequently dictate a larger engine or an alternative power source (e.g., a gas turbine). This is especially true with generator and compressor applications which are the major uses of large-bore engines. Third, improvements in engine design have led to lower fuel consumption in newer engines, which favors replacement of older, less efficient engines given the ever-rising costs of fuels.

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[2] Twenty-thousand hours for large-bore, medium-speed (1200-rpm) spark ignition engines and 80,000 hours for large-bore, low-speed (<400-rpm) compression ignition engines.

Medium-large power (>500 hp but <100 hp/cylinder) diesel-fueled engines also receive minor and major overhauls during their life(25-28). These overhauls, however, are different than overhauls performed on large-bore engines because they are patterned after maintenance practices used by the truck industry. The minor overhauls are termed "kit" overhauls and are performed on the engine where installed. They include replacement of the pistons, rings, liners, valves, and injectors. Major overhauls, on the other hand, require removal of the engine and include replacement or regrinding of the crankshaft and replacement of the main bearings, gears, cylinder heads, and rods. Although major overhauls usually cost less than one-third the price of a new engine(29), the problems of servicing older engines, changing power requirements, and increasing labor costs encourage eventual retirement of an engine.3/ However, since it is current practice to replace substantial portions of these engines, it would be difficult to discriminate between a major overhaul that was performed to avoid the purchase of a new controlled engine from one that was performed in accordance with a routine maintenance program.

## 5.4 SUMMARY

The preceding sections of this chapter have detailed those engine changes which an owner or operator might undertake during its lifetime. It is noted that both minor and major overhauls are performed routinely and usually the old parts are rebuilt to original specifications or replaced with identical new ones. Thus, such overhauls should be exempt from the regulatory consequences of being a modification or reconstruction, despite

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3/The cost of an overhaul would have to be 50 percent of the cost of a new engine for an overhaul to be considered a "reconstruction" in the sense of having to meet a standard of performance(30).

the fact that substantial portions of the engine can be replaced. Alterations are occasionally made to major components, such as pistons, cylinder heads, or turbochargers and aftercoolers, or to the operating parameters, such as timing, air-to-fuel ratio (beyond the manufacturer's specifications or purposeful alteration), or fuel (when not among those specified for use by the manufacturer), and these changes can cause an increase in emissions for which a standard of performance applies. In this case, these changes constitute a modification and would subject the engine to a standard of performance.

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(4) Ibid., Reconstruction.

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# CHAPTER 6

# ALTERNATIVE EMISSION CONTROL TECHNIQUES

This section presents alternative emissions control techniques that can be used to reduce $NO_x$ emissions. Section 6.1 is a brief review of the control systems that were discussed in Section 4.4, indicating which of the techniques are considered technically viable and effective. In Section 6.2, the effectiveness of these techniques (percentage $NO_x$ reduction) and their associated fuel consumption penalties are summarized. Section 6.3 discusses the research and development effort that engine manufacturers estimate is necessary to achieve $NO_x$ reductions from their products.

## 6.1 REVIEW OF CANDIDATE $NO_x$ CONTROL TECHNIQUES

The $NO_x$ emission control techniques discussed in Section 4.4 are summarized in Table 6-1. In general, these techniques require an adjustment of the engine operating conditions or addition of hardware, or a combination of both. Although these techniques are listed separately, they can be used in combination (see Sections 6.2 and 4.4.10). The control techniques described in Table 6-1 can be divided into three groups based on the relative ease with which they can be implemented. This section describes the techniques within each group and indicates which control approaches are considered technically viable. These techniques will form

## TABLE 6-1. SUMMARY OF $NO_x$ EMISSION CONTROL TECHNIQUES FOR RECIPROCATING IC ENGINES

| CONTROL | PRINCIPLE OF REDUCTION | APPLICATION | BSFC INCREASE | COMMENTS -- LIMITATIONS |
|---|---|---|---|---|
| RETARD<br>Injection (CI)[a]<br>Ignition (SI)[b] | Reduces peak temperature by delaying start of combustion during the combustion stroke. | An operational adjustment. Delay cam or injection pump timing (CI); delay ignition spark (SI). | Yes | Particularly effective with moderate amount of retard; further retard causes high exhaust temperature with possible valve damage and substantial BSFC increase with smaller $NO_x$ reductions per successive degree of retard. |
| AIR-TO-FUEL (A/F) RATIO CHANGE | Peak combustion temperature is reduced by off-stoichiometric operation. | An operational adjustment. Increase or decrease to operate on off-stoichiometric mixture. Reset throttle or increase air rate. | Yes | Particularly effective on gas or dual fuel engines. Lean A/F effective but limited by misfiring and poor load response. Rich A/F effective but substantial BSFC, HC, and CO increase. A/F less effective for diesel-fueled engines. |
| DERATING | Reduces cylinder pressures and temperatures. | An operational adjustment, limits maximum bmep[c] (governor setting). | Yes | Substantial increase in BSFC with additional units required to compensate for less power. HC and CO emission increase also. |
| INCREASE-SPEED | Decreases residence time of gases at elevated temperature and pressure. | Operational adjustment or design change. | Yes | Practically equivalent to derating because bmep is lowered for given bhp requirements. Compressor applications constrained by vibration considerations. Not a feasible technique for existing and most new facilities. |
| DECREASED INLET MANIFOLD AIR TEMPERATURE | Reduces peak temperature. | Hardware addition to increase aftercooling or add aftercooling (larger heat exchanger, coolant pump). | No | Ambient temperatures limit maximum reduction. Raw water supply may be unavailable. |
| EXHAUST GAS RECIRCULATION (EGR)<br>External | Dilution of incoming combustion charge with inert gases. Reduce excess oxygen and lower peak combustion temperature. | Hardware addition; plumbing to shunt exhaust to intake; cooling may be required to be effective; controls to vary rate with load. | No if EGR rates not excessive | Substantial fouling of heat exchanger and flow passages; anticipate increased maintenance. May cause fouling in turbocharged, aftercooled engine. Substantial increases in CO and smoke emissions. Maximum recirculation limited by smoke at near rated load, particularly for naturally aspirated engines. |
| Internal<br>valve overlap or retard | Cooling by increased scavenging, richer trapped air-to-fuel ratio. | Operational hardware modification: adjustment of valve cam timing. | Yes | Not applicable on natural gas engine due to potential gas leakage during shutdown. |
| exhaust back pressure | Richer trapped air-to-fuel ratio. | Throttling exhaust flow. | Yes | Limited for turbocharged engines due to choking of turbocompressor. |
| CHAMBER MODIFICATION<br>Precombustion (CI)<br>Stratified charge (SI) | Combustion in ante-chamber permits lean combustion in main chamber (cylinder) with less available oxygen. | Hardware modification; requires different cylinder head. | Yes | 5 to 10 percent increase in BSFC over open-chamber designs. Higher heat loss implies greater cooling capacity. Major design development. |
| WATER INDUCTION | Reduces peak combustion temperature. | Hardware addition: inject water into inlet manifold or cylinder directly; effective at water-to-fuel ratio = 1 (lb $H_2$)/lb fuel). | No | Deposit buildup (requiring demineralization); degradation of lube oil, cycling control problems. |
| CATALYTIC CONVERSION | Catalytic reduction of NO to $N_2$. | Hardware addition: catalytic converter installed in exhaust plumbing or reducing agent (e.g. ammonia) injected into exhaust stream. | No | Catalytic reduction of NO is difficult in oxygen-rich environment. Cost of catalyst or reducing agent high. Little research applied to large-bore IC engines. |

[a] Compression ignition

[b] Spark ignition

[c] bmep -- brake mean effective pressure

the basis for establishing standards of performance for exhaust $NO_x$ emissions.

The first group of controls includes retard, air-to-fuel changes, decreased manifold air temperature, and derating. As discussed in Section 4.4, these techniques, or some combination of them, appear to be the most promising $NO_x$ control approaches based on their demonstrated effectiveness and relative ease of implementation. They generally require only an operational adjustment or resizing of some components (turbocharger, aftercooler, etc.). The exception is derating, which would require additional units to compensate for decreased horsepower. In general, these techniques will increase operating costs because of increases in fuel consumption. Engine manufacturers have also indicated that changes in engine operating conditions may lead to increased maintenance. For example, one manufacturer established that 4 degrees of ignition retard on a dual-fuel engine resulted in a 24 percent decrease in the service life of the exhaust valves (see Section 4.4.2). No other data have been presented, however, that show adverse effects on engine components for the promising control techniques.

As discussed in Section 4.4, the degree of application of any of these techniques will be limited by practical considerations. For large amounts of ignition/injection retard, fuel penalties are large and exhaust temperatures increase. This may cause reduced exhaust valve life or require valve material that can withstand the elevated temperatures (>1200°F).

Similarly, the amount of air-to-fuel (A/F) change is limited by misfiring or detonation. In addition, mutually aspirated engines may experience large increases in fuel consumption particularly as A/F ratio is decreased from stoichiometric.

Manifold air cooling is limited by the ambient air temperature. At maximum ambient conditions, manifold air temperatures with a reasonably cost-effective heat exchanger can practically be reduced (after turbocharging) to 15 to 20°F above the ambient temperature.

The primary limitation of derating is the competitive disadvantage it places on engines that must compete with non-derated models. For example, applications consisting of several engines may require additional units to satisfy total power requirements as a result of derating. Nevertheless, the data in Section 4.4.1 have shown that a small amount (40 percent) of derate can effect a large $NO_x$ reduction in two-stroke engines. Derate, however, is generally less effective in four-stroke engines.

As discussed earlier, manufacturers may apply these control techniques alone, or in combination, to achieve a given emission level. The particular combination and extent of control applied will largely depend on the engine model. As was shown in Section 4.4.10 this approach is effective and can achieve low $NO_x$ emission levels.

The second group of controls includes exhaust gas recirculation (EGR) and combustion chamber modifications. These controls have demonstrated effective $NO_x$ reductions but will require additional development and durability testing. In contrast to the techniques discussed in the first group of controls, EGR will require additional hardware: plumbing to recirculate the exhaust and a heat exchanger to cool the recirculated gas. Although a limited number of EGR tests have shown effective $NO_x$ reductions (see Section 4.4.6), there is concern that the necessity for cooling the recirculated gas will lead to fouling the flow passages of the cooling heat exchanger as well as the engine turbocharger and aftercooler. The tests conducted to date on large-bore engines have been of short duration ( < 100 hours). Thus, longer testing will be necessary to establish the effects that cooling the recirculated gas will have on maintenance practices. One manufacturer of heavy-duty diesel truck engines has rejected EGR because of excessive fouling ( < 200 hours), and another manufacturer believes the application of EGR must be limited to non-aftercooled turbocharged and naturally aspirated engines with full-load EGR cutoff to prevent excessive smoke (>10- percent opacity)(1,2). (EGR has been successfully applied in combination with other techniques on gasoline-fueled engines.) Therefore, large-bore engine manufacturers will require additional testing of this technique to establish its effect on maintenance practices. Nevertheless, this technique is considered a technically viable $NO_x$ control.

Data from smaller-bore diesel engines indicate that combustion chamber shape has a significant effect on $NO_x$ emissions (see Section 4.4.8). However, none of the domestic large-bore engine manufacturers has any experience with new or modified combustion chamber geometries. Manufacturers have estimated that an extensive development program of 3 to 5 years would be required to establish the emissions benefits of such a major engine redesign. Since there is little experience with this control approach, it is not considered a feasible near-term alternative at this time.

The third group of controls includes water induction, increased speed, and catalytic reduction. These techniques have serious technical or cost limitations, have not been shown effective, or have little experimental data to demonstrate their effectiveness. For example, water induction has shown significant $NO_x$ reductions (see Section 4.4.7), but deposit buildup from untreated water, combustion products, and lube oil fouling have led manufacturers to reject this $NO_x$ control approach. With respect to speed changes, limited data indicate that a speed increase of 10- to 15- percent (with a decrease in torque to maintain the same power) results in $NO_x$ reductions. This technique may be effective for electric generation applications because these engines operate at a constant speed. (New units would use different speed exchange gears than installed units). However, this approach could not be applied to gas engine driven compressors because engine speeds are matched with compressor operating speeds, varying as much as 10 percent in the course of normal operation. Since the largest number of new large-bore engines are sold for compressor applications in oil and gas production and transportation, this technique is impracticable. Similarly, limited data exists for catalytic reduction of $NO_x$ emissions from IC engines. As discussed in Section 4.4.9, this approach would be difficult to apply to the oxygen-rich exhausts of large-bore engines and probably would be very expensive (see Section 8.2.3).

Therefore, retard, air-to-fuel changes, derating, manifold air temperature reduction, exhaust gas recirculation, and combinations of these techniques are considered technically viable $NO_x$ control approaches. The following section summarizes the $NO_x$ reductions and fuel penalties associated with these techniques.

## 6.2 EMISSION REDUCTIONS AND FUEL PENALTIES FOR CANDIDATE $NO_x$ CONTROL TECHNIQUES

In general, application of the technically viable $NO_x$ controls is accompanied by increased fuel consumption. The $NO_x$ reductions and fuel penalties for these techniques were discussed in Chapter 4.4 and are summarized in Table 6-2. In practice, an engine user may apply more or less of these controls, either alone or in combination in an effort to reduce $NO_x$ emissions with a minimum increase in fuel consumption. Conclusions based on Table 6-2 are summarized below by fuel type.

### Diesel

The data show that retard alone, or in combination with either inlet air cooling or air-to-fuel changes, is an effective approach for diesel engines achieving $NO_x$ reductions of 28 to 65 percent. EGR also demonstrates a significant reduction in $NO_x$ (33 percent) accompanied by only a 1-percent fuel consumption increase. With the exception of two data points, these $NO_x$ controls result in less than a 10-percent fuel penalty for all diesel engines.

### Dual-Fuel

A number of control techniques prove effective when applied to dual-fuel engines. Retard and air-to-fuel changes register the largest $NO_x$ reductions (up to 73 percent). Fuel penalties are always less than 10 percent and in most cases less than 5 percent.

### Gas

Similarly, several controls or combinations of controls were effective on gas engines. Maximum $NO_x$ reductions ranged from 40 to 80 percent accompanied by fuel penalties of 5 to 12 percent. As discussed in Section 4.4.3, changes in air-to-fuel ratios are particularly effective in

## TABLE 6-2. NO$_x$ REDUCTION AND FUEL CONSUMPTION PENALTIES FOR DIESEL, DUAL-FUEL, AND GAS ENGINES

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| Control Approach | | Engine Fuel Type | | | | | |
|---|---|---|---|---|---|---|---|
| | | Diesel | | Dual Fuel | | Natural Gas | |
| | | % NO$_x$ Reduction | ΔBSFC, % | % NO$_x$ Reduction | ΔBSFC, % | % NO$_x$ Reduction | ΔBSFC, % |
| Derate (D) | (3%) | -- | -- | -- | -- | <20 | 2 |
| | (6%) | -- | -- | -- | -- | <40 | 3 |
| | (10%) | -- | -- | <20 | 4 | -- | -- |
| | (20%) | <20 | 4 | -- | -- | -- | -- |
| | (25%) | 5-23 | 1-5 | 1-33 | 1-7 | 5-90 | 2-12 |
| Retard (R) | (2°) | <20 | 4 | <20 | 3 | -- | -- |
| | (4°) | <40 | 4 | <40 | 1 | <20 | 3 |
| | (8°) | 28-45 | 2-8 | 50-73 | 3-5 | 8-40 | 2-7 |
| Air-to-fuel (A) | (2%) | -- | -- | -- | -- | <20 | 2 |
| | (3%) | -- | -- | <20 | 0 | -- | -- |
| | (5%) | -- | -- | -- | -- | <40 | 7 |
| | (±10%) | 7-8 | 3 | 25-40 | 1-3 | 20-80 | 5-12 |
| Manifold air (M) temperature | (100°F) | 7-15 | 0-2 | 18-37 | 0-1 | 28 | 0 |
| | (107°F) | -- | -- | -- | -- | <20 | 0 |
| | (113°F) | -- | -- | <20 | 1 | -- | -- |
| Internal EGR | | 5 | 2 | -- | -- | <20 | 5 |
| | | -- | -- | -- | -- | 5-35 | 0-8 |
| External EGR | (10%) | <20 | 1 | 20 | 1 | <20 | 0 |
| | | 33 | 1 | -- | -- | 33 | 0 |
| R&M | | <20 | 1 | <20 | 1 | <20 | 3 |
| | | 10-24 | 0-1 | 25 | 2 | 30-40 | 5-6 |
| R&A | | <20 | 8 | <20 | 1 | <20 | 4 |
| | | <40 | 16 | <40 | 2 | <40 | 8 |
| | | 35-65 | 5-26 | 56 | 2 | 17-52 | 4-11 |
| R&M&A | | 20 | 0 | <20 | 2 | <20 | 2 |
| | | -- | -- | 40 | 3 | <40 | 4 |
| | | -- | -- | -- | -- | 40-65 | 6-7 |
| A&M | | <20 | 2 | -- | -- | -- | -- |
| | | 20-30 | 3 | -- | -- | -- | -- |

reducing $NO_x$ from gas engines. This is because $NO_x$ emissions from spark-ignition engines are very sensitive to air-to-fuel ratio. In fact, investigators have shown that derating, retard, and manifold air cooling achieve $NO_x$ reductions from gas engines by, in effect, changing the air-to-fuel ratio(3).

Ranking of Controls

The $NO_x$ reductions summarized in Table 6-2 are reorganized in Table 6-3. The techniques have been grouped into three levels of $NO_x$ reduction: 0 to 20 percent; 20 to 40 percent; and greater than 40 percent. This format readily identifies which control techniques have established given amounts of $NO_x$ reduction. The table indicates that only retard and retard plus air-to-fuel ratio changes achieve $NO_x$ reductions greater than 40 percent for diesel and dual-fuel engines. Several techniques, however, exhibit $NO_x$ reductions greater than 40 percent in natural gas engines. These results are aggregated from a number of engine tests, and not all engines within each fuel category have demonstrated the $NO_x$ reductions shown at each level. Nevertheless, several different techniques applied separately or in combination have been shown effective in achieving significant $NO_x$ reductions (20 to 60 percent).

These techniques are summarized in Table 6-4 for specific levels (20, 40, and 60 percent) of $NO_x$ reduction with the corresponding range of increases in brake-specific fuel consumption. This table illustrates that a number of techniques are capable of achieving a 20-percent $NO_x$ reduction for engines of all three fuels, and fuel penalties are generally less than 5 percent. Fewer controls have been demonstrated for a 40-percent $NO_x$ reduction, and fuel penalties are larger, exceeding 10

TABLE 6-3. RANGES OF $NO_x$ REDUCTION ACHIEVED BY VARIOUS CONTROL TECHNIQUES

| % $NO_x$ Reduction | Diesel | | Dual-Fuel | | Gas | |
|---|---|---|---|---|---|---|
| | Control[a] | ΔBSFC, % | Control[a] | ΔBSFC, % | Control[a] | ΔBSFC, % |
| $\leq 20$ | A, M, IE, RMA, D, R, EE, RM, RA, AM | 1 to 8 | EE, D, R, A, M, RM, RA, RMA | 0 to 4 | D, R, A, M, IE, EE, RM, RA, RMA | 0 to 5 |
| 20 to $\leq 40$ | D, EE, RM, AM, R, RA | 1 to 16 | D, A, M, RM, RMA, R, RA | 1 to 7 | R, M, IE, EE, RM, D, A, RA, RMA | 0 to 8 |
| >40 | R, RA | 8 to 26 | R, RA | 2 to 5 | D, A, RA, RMA | 5 to 12 |
| Maximum reduction (%) | RA (65) | 26 | R (73) | 3 | D (90) | 12 |

[a] R-Retard, M-Manifold air cooling, A-Air-to-fuel change, D-Derate, EE-External EGR, IE-Internal EGR

## TABLE 6-4. $NO_x$ CONTROL TECHNIQUES THAT ACHIEVE SPECIFIC LEVELS OF $NO_x$ REDUCTION

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| $NO_x$ Reduction | Diesel | | Dual Fuel | | Natural Gas | |
|---|---|---|---|---|---|---|
| | Control (amount) | ΔBSFC, % | Control (amount) | ΔBSFC, % | Control (amount) | ΔBSFC, % |
| 20% | Retard (2 to 4°) | 0 to 4 | Retard (2 to 3°) | 1 to 3 | Retard (4 to 5°) | 1 to 4 |
| | External EGR (7%) | 0 | Manifold air cooling | 1 | Manifold air cooling | 0 |
| | Derate (25 to 50%) | 3 to 5 | External EGR (10%) | 1 | External EGR (4%) | 0 |
| | Air-to-fuel change (25%) | 10 | Derate (12 to 25%) | 0 to 8 | Derate (5 to 35%) | 2 to 6 |
| | Retard & manifold air cooling | 0 to 1 | Air-to-fuel changes (5 to 10%) | 0 to 2 | Air-to-fuel change (5%) | 2 |
| | Retard & manifold air cooling & air-to-fuel change | | | | | |
| 40% | Retard (7 to 8°) | 4 to 8 | Retard (5°) | 2 | Retard (10°) | 2 |
| | Derate (50%) | 14 to 17 | Manifold air cooling | 1 | Derate (10 to 50%) | 2 to 24 |
| | Air-to-fuel change & manifold air cooling | 3 to 5 | Derate (30 to 50%) | 2 to 8 | Air-to-fuel change (7%) | 2 |
| | Retard & air-to-fuel change | | Air-to-fuel change (10%) | 2 | Retard & manifold air cooling & air-to-fuel change | 7 |
| | | | Retard & manifold cooling | 3 | | |
| 60% | Retard (16°) | 19 to 24 | Retard (6°) | 2 | Derate (10 to 50%) | 2 to 22 |
| | Retard & air-to-fuel change | 21 | Derate (50%) | 12 | Air-to-fuel change (8 to 12%) | 2 to 5 |
| | | | Retard & air-to-fuel change | 1 to 3 | Retard & manifold air cooling & air-to-fuel change | 7 |

percent in two cases. In general, combinations of retard, air-to-fuel ratio changes, and manifold air cooling were required for engines of any fuel category to achieve a 60-percent $NO_x$ reduction. Fuel penalties are highest for diesel engines, exceeding 20 percent. Fuel penalties are more scattered for dual-fuel and natural gas engines, ranging from 1 to 22 percent but, in general, exceeding 5 percent. The data indicate that retard is the most effective $NO_x$ control for diesel and dual-fuel engines, and that air-to-fuel changes are most effective for dual-fuel and natural gas engines. The cost impact of these alternatives for $NO_x$ emission reductions will be discussed further in Chapter 8.

## 6.3 ESTIMATED TIME TO IMPLEMENT $NO_x$ CONTROL TECHNIQUES

Because manufacturers of large-bore engines are committed to a particular design approach, they conduct extensive research, development, and prototype testing before releasing a new engine model for sale. Consequently, these manufacturers will require some period of time to modify or reoptimize and test engines to meet standards of performance. In general, this requirement consists of time to accomplish the following tasks:

1. Establish baseline (uncontrolled) emissions for all engine models
2. Establish the effectiveness of various techniques on each model and size
3. Perform durability tests on selected engines for certain techniques
4. Engineer any necessary redesigns, including new patterns and tooling

As shown in Section 4.3, engine manufacturers have obtained uncontrolled emissions for about 80 percent of all diesel, dual-fuel, and natural gas engine models. Furthermore, engine manufacturers have expended considerable effort to investigate potential $NO_x$ emission controls. Therefore, the time requirements for areas 1 and 2 will be primarily to investigate the effectiveness of applying a given control to a particular engine model to meet a specified emission level.

Manufacturers have also indicated that a minimum durability test of 2000 hours is required for any engine model whose operating or combustion characteristics are modified. For major design changes, on the other hand (e.g., combustion chamber redesign), they estimated a need for 8000 hours (about 1 year) of testing.

Finally, control techniques that would require special hardware and/or modification of existing engine hardware would involve some engineering and pattern and tooling efforts as part of their development. EGR and combustion chamber redesign are techniques that would require this development.

Table 6-5 estimates the time required to accomplish the four tasks for all of the candidate control techniques identified in Section 6.1. The entries in this table are based on the following factors:

- An estimate of 9 months is required for all engines that need controls. Since not all manufacturers own their own emission measurement equipment (see Section 4.2.4) or have data on the emissions of their current production engines, 6 months have been allowed for the procurement of this equipment, staff training, and the measurement of baseline emission data on a variety of engine models. An additional 3 months are allotted

TABLE 6-5. ESTIMATED TIME SPAN BETWEEN DECISION TO CONTROL AND START OF PRODUCTION OF FIRST CONTROLLED ENGINE (IN MONTHS)

| Control Technique | Baseline Testing and Control Selection[a] | Feasibility and Development | Control Effectiveness and Durability Testing | Production Retooling | Total Time |
|---|---|---|---|---|---|
| Derating | 9 | 0 | 0 | 0 | 9 |
| Retard | 9 | 0 | 6 | 0 | 15 |
| Air-to-fuel | 9 | 0 | 6 | 0 | 15 |
| Manifold air temperature | 9 | 0 | 6 | 0 | 15 |
| EGR, external | 9 | 8 | 12 | 6 | 35 |
| EGR, internal | 9 | 4 | 8 | 4 | 25 |
| Combustion chamber redesign | 9 | 36 | 12 | 12 | 69 |

[a] Estimate includes 6 months of baseline engine testing plus 3 months for control selection decision.

to analyze the baseline results, consider the effectiveness of the various controls available for each engine, and decide on a control strategy.

- If problems are encountered during the long-term durability tests, the problems need to be corrected and some of the tests rerun. Therefore, 12 months have been allowed for durability testing of major changes. Since internal EGR is not as significant a change as external EGR or a chamber redesign, only 8 months were allotted for durability testing of this technique. Also, since there is no reason to expect maintenance or durability problems with an engine that is operated at less than full capacity, no time is allocated to durability testing of a derated engine.
- Retard and air-to-fuel ratio adjustments require no new hardware and could be implemented after durability testing. For EGR (internal and external) the time to assemble new production equipment was scaled down from the comparable time for a combustion chamber redesign, again after considering the relative complexities of the various techniques.
- The time to develop a combustion chamber redesign for a large engine was estimated for large-bore engine manufacturers at 3 to 3-1/2 years, exclusive of durability testing and time to assemble the necessary production tooling(4,5). This estimate, in one case, is for a redesign of the valves, injectors, and piston head and is based on the use of twice the manufacturers's current R&D resources for this effort(5). Development times for modified injectors, EGR, and water

induction were estimated by considering the potential technical difficulty with these systems relative to those of modified chambers.

In summary, changes to engine operating conditions or the addition of existing hardware could be implemented in 9 to 15 months, and a major engine combustion system redesign would take 5 to 6 years.

## REFERENCES FOR CHAPTER 6

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# CHAPTER 7

# ENVIRONMENTAL IMPACT

This chapter considers the environmental impacts, both beneficial and adverse, associated with the control of exhaust emissions from stationary reciprocating internal combustion (IC) engines. The candidate emission control systems are discussed in Chapter 4 and are designed primarily to reduce $NO_x$ emissions from large-bore stationary IC engines. The major emphasis of this chapter will be to identify the incremental impact as compared to uncontrolled engines of these emission control systems on air, water, solid waste, noise pollution, and energy consumption. In addition, secondary effects resulting from these control systems (e.g., increased HC or CO emissions as a consequence of reducing $NO_x$ emissions) will be identified. The incremental impact of these systems will be related to uncontrolled engines since virtually no stationary IC engines are currently subject to emission regulations other than for smoke.

## 7.1 AIR POLLUTION IMPACT

The discussion in Section 9.1 indicates that currently installed stationary IC engines contributed approximately 8.4 percent of the total U.S. emissions of $NO_x$. Furthermore, large-bore engines (greater than 350 CID/cyl) contribute approximately three-fifths of the $NO_x$ emissions from stationary IC engines, or nearly 5 percent of the nationwide burden.

The projected impact of a standard for large-bore stationary engines on nationwide emissions will be addressed in Section 7.6. This section will evaluate the incremental impact of the candidate $NO_x$ emission control systems (identified in Chapter 6) on ambient air quality. To do this, ambient air concentrations of $NO_x$, CO, HC, $SO_x$, and particulates resulting from emissions from stationary engines will be estimated using an atmospheric dispersion model. Calculations will be performed for both uncontrolled and controlled engine emissions under adverse meteorological conditions. These concentrations will then be compared with corresponding ambient air quality standards. The following sections will briefly discuss model plant characteristics, the atmospheric dispersion model used, and the results of the calculations.

### 7.1.1 Model Plant Characteristics

Data presented in Section 4.3 indicate the following range of uncontrolled emissions for large-bore IC engines:

|  | Emissions, g/hp-hr |  |  |
|---|---|---|---|
|  | $NO_x$ | CO | $HC_T$ |
| Diesel | 5 - 18 | 0.5 - 8.5 | 0.1 - 1.0 |
| Dual Fuel | 4 - 13 | 1.2 - 4.5 | 0.8 - 5.7 |
| Gas | 8 - 29 | 0.2 - 29 | 0.3 - 5.8 |

In general, CO is quite low in comparison with controlled mobile diesel and gasoline engine emissions (the current federal CO standard for heavy-duty vehicle diesel engines is 40 g/hp-hr; the California 1977 standard is 25 g/hp-hr). Total hydrocarbon emissions for large-bore diesel engines are about one-half those from currently controlled heavy-duty vehicle diesel engines. Also, these levels are about one-half the levels that will be emitted when automotive (light-duty vehicle) engines are controlled to comply with the ultimate statutory limit of 0.41 grams per mile specified in the 1970 and 1977 Amendments to the Clean Air Act.

Four typical large-bore engine configurations were chosen for the atmospheric dispersion calculations. In addition, a compressor station was simulated by modeling the dispersion from a row of seven 1500-hp engines. Table 7-1 presents the physical dimensions and emission source characteristics of all these engines. The $NO_x$ emission rates are given for sales-weighted uncontrolled levels (see Section 4.3.4) and for three levels of control corresponding approximately to 20, 40 and 60 percent $NO_x$ reductions from sales-weighted uncontrolled levels. The CO and $HC_T$ levels correspond to values that most engines can achieve when controlled for $NO_x$. (See Figures 4-52 and 4-53.)

An unfavorable feature of the designs of the five prototypes is that the height of the exhaust discharge is not significantly greater than surrounding structures. This causes aerodynamic complications which can seriously interfere with the rise of the effluent plume, thereby producing significantly higher ground-level concentrations.

### 7.1.2 The Dispersion Model and Meteorological Considerations

The dispersion model used to analyze these plants is the single source model developed by the Meteorology Laboratory, EPA. The dispersion analysis was performed by Walden Research Division of Abcor, Inc. under contract to the Monitoring and Data Analysis Division of EPA(1).

In this model all pollutants are assumed to display the dispersion behavior of nonreactive gases. The predicted pollutant concentrations

## TABLE 7-1. EMISSION SOURCE CHARACTERISTICS OF MODEL STATIONARY INTERNAL COMBUSTION ENGINE INSTALLATIONS

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| Engine Characteristics | | | Height of exhaust discharge above ground level, meters | Building height meters | Inside diameter of exhaust discharge, meters | Exhaust temperature at discharge, °K | Exhaust gas vertical velocity, meters/seconds | Control level[a] | Emission Rate | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Rated Horsepower | Rated rpm | Fuel | | | | | | | $NO_x$[a] | | CO[c] | | $SO_2$[b] | | $HC_T$[c] | | Particulates | |
| | | | | | | | | | g/hp-hr | g/sec | g/hp-hr | g/sec | g/hp-hr | g/sec | g/hp-hr | g/sec | mg/scf | g/sec |
| 1200 | 335 | Diesel | 10.0 | 8.0 | 0.61 | 700 | 14.3 | U | 11 | 3.69 | 5.0 | 1.68 | 0.92 | 0.31 | 0.4 | 0.13 | 0.5 | 0.075 |
| | | | | | | | | $C_1$ | 9 | 2.95 | -- | -- | -- | -- | -- | -- | -- | -- |
| | | | | | | | | $C_2$ | 7 | 2.21 | -- | -- | -- | -- | -- | -- | -- | -- |
| | | | | | | | | $C_3$ | 4 | 1.48 | -- | -- | -- | -- | -- | -- | -- | -- |
| 4300 | 600 | Dual fuel | 10.0 | 8.0 | 0.61 | 700 | 14.3 | U | 8 | 9.51 | 5.0 | 6.00 | -- | -- | 6.0 | 7.15 | -- | -- |
| | | | | | | | | $C_1$ | 6 | 7.61 | -- | -- | -- | -- | -- | -- | -- | -- |
| | | | | | | | | $C_2$ | 5 | 5.71 | -- | -- | -- | -- | -- | -- | -- | -- |
| | | | | | | | | $C_3$ | 3 | 3.80 | -- | -- | -- | -- | -- | -- | -- | -- |
| 1500 | 250 | Natural gas | 10.0 | 8.0 | 0.91 | 700 | 13.4 | U | 15 | 6.3 | 1.0 | 0.42 | -- | -- | 5.0 | 2.10 | -- | -- |
| | | | | | | | | $C_1$ | 12 | 5.0 | -- | -- | -- | -- | -- | -- | -- | -- |
| | | | | | | | | $C_2$ | 9 | 3.8 | -- | -- | -- | -- | -- | -- | -- | -- |
| | | | | | | | | $C_3$ | 6 | 2.5 | -- | -- | -- | -- | -- | -- | -- | -- |
| 4400 | 327 | Natural gas | 10.0 | 8.0 | 0.91 | 700 | 13.4 | U | 15 | 18.0 | 1.0 | 1.20 | -- | -- | 5.0 | 6.09 | -- | -- |
| | | | | | | | | $C_1$ | 12 | 14.4 | -- | -- | -- | -- | -- | -- | -- | -- |
| | | | | | | | | $C_2$ | 9 | 8.4 | -- | -- | -- | -- | -- | -- | -- | -- |
| | | | | | | | | $C_3$ | 6 | 7.2 | -- | -- | -- | -- | -- | -- | -- | -- |
| Row of 7 1500-hp engines to simulate a compressor station | 250 | Natural gas | 10.0 | 8.0 | 0.91 | 700 | 13.4 | U | 15 | 39.6 | 1.0 | 2.93 | -- | -- | 5.0 | 14.68 | -- | -- |
| | | | | | | | | $C_1$ | 12 | 31.7 | -- | -- | -- | -- | -- | -- | -- | -- |
| | | | | | | | | $C_2$ | 9 | 23.8 | -- | -- | -- | -- | -- | -- | -- | -- |
| | | | | | | | | $C_3$ | 6 | 15.8 | -- | -- | -- | -- | -- | -- | -- | -- |

[a] Emission rates are: U -- Sales-weighted average uncontrolled levels
$C_1$, $C_2$, $C_3$ -- 20-, 40-, and 60-percent reduction $NO_x$, respectively, from average uncontrolled levels (to the nearest 1 g/hp-hr).

[b] Based on diesel fuel with 0.22 percent sulfur by weight and complete conversion of all fuel sulfur to $SO_2$.

[c] CO and $HC_T$ levels represent values that most engines can achieve when controlled for $NO_x$ (See Figures 4-52 and 4-53).

are based on the application of state-of-the-art modeling techniques, which implies reliability of the estimates to within a factor of about two(2).

The following assumptions are applied in the analytical approach:

1. There are no significant seasonal or hourly variations in emission rates for these plants
2. The plants are located in flat or gently rolling terrain. In restrictive terrain, the dispersion of effluents could be more impaired, resulting in higher ambient concentration levels
3. The meteorological regime is unfavorable to the dispersion of effluents
4. All engines are continuously in operation. The 24-hour and annual ambient concentrations for engines which do not operate continuously will likely be lower

Preliminary analyses indicated that, for the stationary internal combustion engines, critical meteorological conditions (i.e., those giving rise to maximum short-term impact) consist of a combination of stable atmospheric conditions and moderate wind speed. If such conditions occur frequently at a given location, especially if they are combined with a high directional bias in the wind, then longer-term impact (e.g., 24 hours and annual) will also tend to be high.

Stationary internal combustion engines also have a geographical tendency which must be considered. A large portion of these engines are located in the midwestern to southwestern United States. Considering these factors, Oklahoma City was chosen as the "worst case" location.

The model is programmed to use a previously determined set of dispersion conditions derived from the basic meteorological data for each hour of the given year. The calculations simulate the interaction between the plant characteristics and these dispersion conditions to produce a dispersion pattern for each hour. The computations are performed for each point in an array of 180 receptors for any number of hours. In the case of stationary internal combustion engines, the averaging periods of interest are 1 hour, 3 hours, 8 hours, 24 hours, and annual, depending on the pollutant of interest.

The prototype stationary internal combustion engines are modeled with aerodynamic effects taken into consideration -- i.e., with a computational procedure that accounts for the flowfield interactions between the plume wind, and buildings. Such effects were found to be critical due to unfavorable design (i.e., the exhaust discharge location is not significantly higher than the surrounding buildings).

### 7.1.3 Results and Discussion of Dispersion Calculations

The results of the dispersion calculations are presented in Table 7-2. Maximum short-term average (CO, $SO_x$, HC, and particulates) and annual average ($NO_x$) concentrations (at a distance of 0.3 kilometers from the engine exhaust stack) are presented for the levels of emissions listed in Table 7-1. Levels for CO and $HC_T$ that were used in the dispersion calculation are typical of controlled engines (see Figures 4-52 and 4-53). The potential environmental gain from a regulation based on the lowest achievable levels of CO and $HC_T$ are discussed in Subsection 7.1.5 below. The last column in Table 7-2 indicates the percent of hourly concentrations that are greater than 50 percent of the maximum concentration during a 1-year period. National primary and secondary ambient air quality standards for these pollutants are presented in Table 7-3(3) for comparison with the results in Table 7-2.

TABLE 7-2. ESTIMATED GROUND LEVEL CONCENTRATIONS (at 0.3 km from source) DUE TO EMISSIONS FROM STATIONARY INTERNAL COMBUSTION ENGINES[a]

| Engine Characteristics[b] | Pollutant[c] | Control[d] Level | Emission Rate | | Averaging Period | Maximum Concentrations, µg/m³ | % of Concentrations Above 50% Maximum Over 1-Year Period |
|---|---|---|---|---|---|---|---|
| | | | g/hp-hr | g/s | | | |
| 1200 hp, 835 rpm<br>Diesel Fuel | $NO_x$ | U<br>$C_1$<br>$C_2$<br>$C_3$ | 11<br>9<br>7<br>4 | 3.69<br>2.95<br>2.21<br>1.48 | Annual<br>Annual<br>Annual<br>Annual | 61<br>50<br>39<br>22 | --<br>--<br>--<br>-- |
| | CO | C | 5.0 | 1.68 | 1 hour<br>8 hours | 1.4 mg/m³<br>0.4 mg/m³ | <1%<br>9% |
| | $SO_2$ | U | 0.93 | 0.31 | 3 hours<br>24 hours<br>Annual | 120<br>57<br>5.2 | 3%<br>30%<br>-- |
| | $HC_T$ | C | 0.4 | 0.13 | 3 hours | 50 | 3% |
| | Particulate | U | 0.22 | 0.075 | 24 hours<br>Annual | 6.6<br>1.3 | 30%<br>-- |
| 4300 hp, 600 rpm<br>Dual Fuel | $NO_x$ | U<br>$C_1$<br>$C_2$<br>$C_3$ | 8<br>6<br>5<br>3 | 9.51<br>7.61<br>5.71<br>3.80 | Annual<br>Annual<br>Annual<br>Annual | 160<br>120<br>100<br>60 | --<br>--<br>--<br>-- |
| | CO | C | 5.0 | 6.0 | 1 hour<br>8 hours | 5 mg/m³<br>1.2 mg/m³ | <1%<br>9% |
| | $HC_T$ | C | 6.0 | 7.15 | 3 hours | 2,768 | 3% |
| | NMCH | C | -- | -- | 3 hours | (est.)692 | |

[a] All maximum concentrations are located at 0.3 km from the source

[b] Rated power and speed

[c] $NO_x$ in µg $NO_2/m^3$; NMHC (nonmethane HC) estimated to be 10% of $HC_T$ (total HC) for gas, 25% for dual fuel, and 100% for diesel engines

[d] Emission rates are: U -- sales weighted uncontrolled levels
$C_1$, $C_2$, $C_3$ -- 20, 40, and 60% respective reduction of $NO_x$ from average uncontrolled levels (to the nearest 1 g/hp-hr)
C -- levels typical of controlled engines

## TABLE 7-2. Continued

| Engine Characteristics[b] | Pollutant[c] | Control[d] Level | Emission Rate |  | Averaging Period | Maximum Concentrations, µg/m$^3$ | % of Concentrations Above 50% Maximum Over 1 Year Period |
|---|---|---|---|---|---|---|---|
|  |  |  | g/hp-hr | g/s |  |  |  |
| 1500 hp, 250 rpm<br>Natural Gas Fuel | NO$_x$ | U<br>C$_1$<br>C$_2$<br>C$_3$ | 15<br>12<br>9<br>6 | 6.3<br>5.0<br>3.8<br>2.5 | Annual<br>Annual<br>Annual<br>Annual | 73<br>58<br>44<br>29 | --<br>--<br>--<br>-- |
|  | CO | C | 1.0 | 0.42 | 1 hour<br>8 hours | 0.1 mg/m$^3$<br><0.1 mg/m3(0.036) | 8%<br>7% |
|  | HC$_T$ | C | 5.0 | 2.1 | 3 hours | 474 | 8% |
|  | NMHC | C | -- | -- | 3 hours | (est.) 47 | 8% |
| 4400 hp, 327 rpm<br>Natural Gas Fuel | NO$_x$ | U<br>C$_1$<br>C$_2$<br>C$_3$ | 15<br>12<br>9<br>6 | 18.0<br>14.4<br>8.4<br>7.2 | Annual<br>Annual<br>Annual<br>Annual | 223<br>178<br>104<br>89 | -<br>-<br>-<br>-- |
|  | CO | C | 1.0 | 1.2 | 1 hour<br>8 hours | 0.4 mg/m$^3$<br>0.16 mg/m$^3$ | 8%<br>7% |
|  | HC$_T$ | C | 5.0 | 6.09 | 3 hours | 1414 | 8% |
|  | NMHC | C | -- | -- | 3 hours | (est.) 141 | 8% |

[a] All maximum concentrations are located at 0.3 km from the source

[b] Rated power and speed

[c] NO$_x$ in µg NO$_2$/m$^3$; NMHC (nonmethane HC) estimated to be 10% of HC$_T$ (total HC) for gas, 25% for dual fuel, and 100% for diesel engines

[d] Emission rates are: U -- sales weighted uncontrolled levels
C$_1$, C$_2$, C$_3$ -- 20, 40, and 60% respective reduction of NO$_x$ from average uncontrolled levels (to the nearest 1 g/hp-hr)
C -- levels typical of controlled engines

## TABLE 7-2. Concluded

| Engine Characteristics[b] | Pollutant[c] | Control[d] Level | Emission Rate |  | Averaging Period | Maximum Concentrations, µg/m³ | % of Concentrations Above 50% Maximum Over 1 Year Period |
|---|---|---|---|---|---|---|---|
|  |  |  | g/hp-hr | g/s |  |  |  |
| Compressor Station<br>7 Engines, each 1500 hp<br>250 rpm<br>Natural Gas Fuel | $NO_x$ | U<br>$C_1$<br>$C_2$<br>$C_3$ | 15<br>12<br>9<br>6 | 39.6<br>31.7<br>23.8<br>15.8 | Annual<br>Annual<br>Annual<br>Annual | 511<br>409<br>307<br>204 | -<br>-<br>-<br>- |
|  | CO | C | 1.0 | 2.93 | 1 hour<br>8 hours | 0.95 mg/m³<br>0.57 mg/m³ | 8%<br>7% |
|  | $HC_T$ | C | 5.0 | 14.62 | 3 hours | 3,311 | 8% |
|  | NMHC | C |  |  | 3 hours | (est.) 331 | 8% |

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[a] All maximum concentrations are located at 0.3 km from the source

[b] Rated power and speed

[c] $NO_x$ in µg $NO_2/m^3$; NMHC (nonmethane HC) estimated to be 10% of $HC_T$ (total HC) for gas, 25% for dual fuel, and 100% for diesel engines

[d] Emission rates are: U -- sales weighted uncontrolled levels
$C_1$, $C_2$, $C_3$ -- 20, 40, and 60% respective reduction of $NO_x$ from average uncontrolled levels (to the nearest 1 g/hp-hr)
C -- levels typical of controlled engines

TABLE 7-3. NATIONAL AMBIENT AIR QUALITY STANDARDS(3)

| Pollutant | Averaging Time and Method | Standard, $\mu g/m^3$ | |
|---|---|---|---|
| | | Primary | Secondary |
| $NO_2$ | Annual arithmetic mean | 100 | 100 |
| NMHC[a] | 3 hr: 6 - 9 a.m. | 160 | 160 |
| CO | 8 hr | 10 $mg/m^3$ | 10 $mg/m^3$ |
| | 1 hr | 40 $mg/m^3$ | 40 $mg/m^3$ |
| $SO_2$ | Annual arithmetic mean | 80 | 60 |
| | 24 hr | 365 | 260[b] |
| | 3 hr | - | 1300 |
| Particulate | Annual arithmetic mean | 75 | 60 |
| | 24 hr | 260 | 150 |

[a] Nonmethane hydrocarbons

[b] As a guide to assess implementation for the annual standard

[c] As a guide to assess implementation for the 24-hr standard

All maximum concentrations for the model stationary internal combustion engine are noted at extremely close-in distances (0.3 km). This is due to the aerodynamic effects on plume rise as well as to the relatively low height of the exhaust discharge. These concentrations are estimated to fall off with greater distance at the following rates:

| Distance from source, km | Percent of the 0.3-km level |
|---|---|
| 0.6 | 33 |
| 1.0 | 15 |
| 2.0 | 5 |
| 5.0 | 1.5 |

The following subsections summarize the comparison of the computed concentrations (Table 7-2) with the ambient air quality standards (Table 7-3).

### 7.1.4 $NO_x$ Concentrations

The 1200-hp diesel and 1500-hp gas engines are within ambient air quality standards at both uncontrolled and controlled emission levels when used individually. The larger gas and dual-fuel engines (4300 and 4400 hp), however, can exceed the $NO_x$ standard at their uncontrolled and controlled levels but satisfy this ambient standard when their uncontrolled $NO_x$ emissions are reduced 60 percent. The group of seven 1500-hp engines (typical of a pipeline compressor station) may exceed the standard at the uncontrolled level and the controlled levels. Thus, these large engines could cause the ambient air quality criteria for $NO_x$ to be violated, at a distance where public exposure is probable, even if emissions were reduced significantly by controls. These violations would occur during the adverse meteorological conditions postulated for this analysis. At typical uncontrolled levels (15 g/hp-hr), the ambient air criteria would be met at all receptors which are further than about 1.0 km from the compressor station. The comparable distance would be less than 0.5 km if the seven engines were all controlled to the lowest level indicated.

### 7.1.5 CO and HC Concentrations

Ambient air concentrations presented in Table 7-2 for CO are based on representative levels of CO that all engines for which data are available could achieve when controlled for $NO_x$ emissions. These calculations indicated that none of the engines exceed ambient air quality criteria for CO. This is not unexpected, considering the relatively low emissions of CO from these large-bore engines, which are designed to run at optimum thermal efficiency.

The HC input data to the model were based on representative levels of total hydrocarbons, including methane, that nearly all engines were shown to achieve when controlled for $NO_x$. Measurements by researchers indicate that no more than 10 percent of the total hydrocarbon emissions from gas engines and 25 percent of the total hydrocarbons from dual-fuel engines are reactive HC compounds(4,5). When the concentrations reported in Table 7-2 are multiplied by these fractions, only the 4300-hp dual-fuel engine and the compressor station caused the nonmethane hydrocarbon standard to be exceeded. Within 0.6 km of the station, the ambient air would not exceed the nonmethane standard if the station were the only source. Moreover, if these gas engines at the compressor station are assumed to emit total hydrocarbons at a level of 2.0 g/hp-hr, as achieved by some gas engines, instead of the upper limit value of 5.0 g/hp-hr used for these calculations, then this station, by itself, would not cause the nonmethane hydrocarbon standard to be exceeded.

Although some $NO_x$ emission control technologies cause HC and/or CO emissions to increase, the data in Figures 4-52 and 4-53 show that these increases are generally not large. Thus, for diesel engines the $NO_x$ reductions were generally achieved with HC and CO increased less than 0.2 and 2 g/hp-hr, respectively (from uncontrolled levels of 0.1 to 1.0 g/hp-hr for HC and 1 to 8 g/hp-hr for CO). Similarly, for dual-fuel engines the increases for both HC and CO are usually less than 2 g/hp-hr (from uncontrolled levels of 1 to 6 g/hp-hr), while for gas engines they tend to be about 1 g/hp-hr (from similar uncontrolled levels). Carbureted or naturally aspirated gas engines, however, experience greater increases in HC and CO, since they operate closer to stoichiometric conditions.

### 7.1.6 $SO_x$ Concentrations

The $SO_x$ emission rate listed in Table 7-1 for the diesel engine corresponds to a fuel sulfur content of about 0.22 percent by weight. This level is typical for distillate fuels. By comparison, 0.5 percent is the maximum allowable fuel sulfur content in several states (e.g., Connecticut, Massachusetts (outside of Boston), and Rhode Island). Other areas restrict the fuel to a sulfur content no more than 0.3 percent (e.g., Philadelphia, Boston, and New Jersey). Since the predicted ground level concentrations shown in Table 7-2 are far below both the primary and secondary ambient air standards, they would still be well below these standards even if the fuel contained 0.5 percent sulfur. However, if one scales these computed results directly on emission rate (i.e., if one assumes that doubling either the fuel sulfur content or the engine size results in double the ground level concentration), any engine larger than 5700 hp could cause a violation in the secondary ambient air quality standard (the 3-hour standard is exceeded first in this case) if it burned fuel with 0.5 percent sulfur and if it were the only source.

### 7.1.7 Particulates

Particulates are generally only a problem with diesel engines. As one can see by comparing Table 7-2 with Table 7-3, the model diesel engine does not, by itself, cause the local air to exceed the ambient particulate standard. Even the largest diesel engine produced in the U.S., a 13,500-hp unit, would just meet the primary annual and the secondary 24-hour standard if it were the only contributing source.

## 7.2 WATER POLLUTION IMPACT

The promulgation of standards of performance should have no significant impact on water pollution. Only two control techniques could result in an additional discharge of water -- increased manifold air cooling and water induction. However, since most newly installed engines use a closed cooling system(6), similar to an automobile radiator, the increased cooling would be obtained by means of a larger radiator, a higher speed fan, or both. Even where an open cooling system is used -- e.g., a once-through cooling tower -- no additional water would be discharged to streams, rivers, lakes, or other groundwater systems. The major impacts from a cooling tower are airborne mist and the demand for makeup water.

If water induction were chosen by an engine manufacturer or user as a means of meeting a potential standard, a demand would be created for water, and there would be an airborne discharge of additional water vapor with the engine exhaust. More importantly, however, the inducted water would require treatment to reduce total dissolved solids (TDS -- see Section 4.4.7). One researcher who investigated the effects of water induction on emissions and engine components indicated that the water available to him, which contained approximately 400 ppm of calcium bicarbonate, was unacceptable due to the formation of combustion chamber deposits during daily operation(7).

Although raw water sources and municipal water quality vary considerably, typical values are 64 mg/ℓ TDS for municipal water, and they should typically not exceed 500 mg/ℓ (~500 ppm) of TDS(8). Any water that is to be withdrawn for treatment and distribution as a portable supply in the Ohio River Valley Water Sanitation District, cannot exceed TDS 750 mg/ℓ (at any time). The corresponding regulation for industrial water supplies (industrial cooling and processing) is 100 mg/ℓ TDS(9). Therefore, depending on the source of water to be inducted into the engine, the user may need to treat water ranging from less than 100 mg/ℓ TDS to several hundred mg/ℓ TDS before inducting the water into his engine to reduce $NO_x$.

The wastes associated with this treatment would be required to meet local regulations before discharge to the environment. According to one source(10), the supplier of deionized water or a deionization system assumes the responsibility of waste treatment and disposal of solids associated with the water "softening" process. This disposal problem is not new to the engine industry: cooling water for large-bore engines is presently treated to reduce TDS which would otherwise accumulate as scale and reduce cooling efficiency. According to the above source, users presently neutralize their waste steams and then drain the treated water observing any local regulations that are in effect. Typical treatment and disposal costs are included in the costs of a water treatment facility as discussed in Section 8.2.3. If a 10,000-hp engine in continuous operation (8000 hours per year) used water induction at 400 ppm TDS and a water-to-fuel ratio of 1.0, it would add 5.8 metric tons/yr of solids to the stream of a waste disposal facility.

## 7.3 SOLID WASTE DISPOSAL IMPACT

The demonstrated control technologies discussed in Chapter 6 should also have no impact on solid waste disposal. Some control techniques (e.g., EGR or water induction) may require more frequent oil changes as a result of oil contamination. According to one operator of large-bore engines(11), lubricating oil is changed every 40,000 to 60,000 hours, usually during a major overhaul. In between, the oil is checked about once a month for water and fuel content. Oil is added periodically to replace that lost through blowby and leakage. Water communication necessitates an oil filter change (normally changed at approximately 500-hour intervals) due to damage of the disposal paper element by the water. Contamination of the lubricant by the more volatile fuel requires an oil change because of an explosion hazard. There is no sludge, however, associated with oil filtering, and the discarded oil can be sold to an oil reclaimer for $0.05/gallon or burned as a waste fuel in a boiler.

## 7.4 ENERGY IMPACT

The annual incremental impact on national energy consumption can be estimated based on historical production data (see Section 9.2) and fuel penalties for candidate $NO_x$ emission controls (see Section 6.3). Table 7-4(12) presents the annual incremental energy impact after 5 and 10 years of a standard of performance. A 10-percent fuel penalty was assumed for all new sales during this period. This assumption is conservative since many techniques discussed in Chapter 6 achieved large $NO_x$ reductions with less than a 10-percent fuel penalty. Thus, this evaluation is an

## TABLE 7-4. ESTIMATED ANNUAL ENERGY IMPACT OF LARGE-BORE ENGINES

| Fuel | Number units added annually | Added hp x10³ | Usage hr/yr | Brake specific fuel consumption Btu/hp-hr | Fuel penalty percent | (Annual increased fuel consumption) | | (5th year increased fuel consumption) | | | (10th year increased fuel consumption) | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | (Btu/yr) x 10¹² | Quantity[c] | (Btu/yr) x 10¹² | Quantity[c] | Percent of 1972 total[b] | (Btu/yr) x 10¹² | Quantity[c] | Percent of 1972 total[b] |
| Diesel | 25 | 87 | 8000 | 7000 | 10 | 0.487 | 84,000 bbl | 2.44 | 420,000 bbl | 0.0044 | 4.87 | 840,000 bbl | 0.0087 |
| Dual fuel[a] | 12 | 30 | 8000 | 6500 | 10 | 0.156 | $0.156 \times 10^9$ scf | 0.78 | $0.78 \times 10^9$ scf | 0.0014 | 1.56 | $1.56 \times 10^9$ scf | 0.0028 |
| Natural gas | 400 | 524 | 8000 | 7500 | 10 | 3.14 | $3.14 \times 10^9$ scf | 15.7 | $15.7 \times 10^9$ scf | 0.0280 | 31.4 | $31.4 \times 10^9$ scf | 0.0560 |
| TOTAL | 437 | 641 | -- | -- | -- | 3.783 | -- | 18.92 | -- | 0.0340 | 37.83 | -- | 0.0670 |

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[a] Assumed 100 percent natural gas

[b] Annual U.S. consumption in 1972 (Reference 12)

Natural gas: $23,125 \times 10^{12}$ Btu
Diesel oil: $32,965 \times 10^{12}$ Btu
Total $56,090 \times 10^{12}$ Btu

Total annual impact: $(3.783/56,090) \times 100 = 0.0067\%$

[c] bbl -- barrels of oil
scf -- standard cubic feet of gas (1000 Btu/scf)

upper estimate based on an unusually high fuel penalty, and it is not intended to apply to a specific emission level or engine and control combination. Despite this assumption, Table 7-4 shows that the application $NO_x$ controls would cause less than a 0.07-percent increase in nationwide consumption of oil and gas after 10 years.

Annual production shown in Table 7-4 is computed as an historical 5-year sales average based on information presented in Section 8.1. These data show that sales of stationary engine horsepower that would be affected by a standard of performance have been essentially constant over the past 5 years and are assumed to remain relatively constant in the future. Applications of these affected engines include electric generation for diesel and dual-fuel engines, and oil and gas production or transportation for natural gas engines. All of these applications are high usage, estimated as 8000 hours per year. Typical fuel consumption values are based on the data from Figure 4-13.

These assumptions result in an annual increase of domestic oil and gas consumption of 0.0067 percent relative to 1972 total consumption. This penalty would increase to 0.034 percent after 5 years and 0.067 percent after 10 years, again relative to the 1972 total consumption. If, however, total consumption were assumed to increase by 3.5 percent per year, the effect of a 10-percent fuel penalty would cause the increase in domestic oil and gas consumption to be only 0.029 percent of the nation's fuel consumption 5 years hence, and 0.048 percent of the total domestic usage in 10 years.

## 7.5 OTHER ENVIRONMENTAL IMPACTS

### 7.5.1 Noise Impact

Fan noise levels from large-bore stationary reciprocating engine installations could increase as a result of increased cooling requirements if decreased manifold air temperature or precombustion chambers (which have higher heat rejection) are used to reduce emissions. However, in typical installations such as municipal generator plants, pipeline compressor stations, or industrial process plants, such increases in the noise level are likely to be insignificant in comparison to other sources of noise (e.g., generators, compressors, pumps, or process blowers).

Although stationary engines are not specifically regulated noise sources, some engine installations mentioned above may be subject to Occupational Safety and Health Act (OSHA) regulations or EPA guidelines regarding noise exposure limits for employees. Allowable noise levels and employee exposure times are specified in parts 55 and 56 of the August 7, 1974, Federal Register, Volume 39, No. 153. In any new installation, measures should be taken as required to meet these regulations.

### 7.5.2 Thermal Pollution Impact

The application of emission controls can increase fuel consumption with no corresponding increase in power. This additional heat input is discharged to the environment as waste heat either by the cooling system or with the exhaust gases. In an attempt to identify the increment of thermal pollution caused by emission control techniques, two model engines are considered below.

Figure 4-51 illustrates that the range of fuel penalties associated with emission control techniques for large-bore engines is about 1 to

10 percent. The following data indicates, as a worst case example, the additional heat discharged to the atmosphere as a result of a 10-percent fuel penalty for both a 500-hp and a 13,500-hp engine in continuous operation (800 hr) for 1 year. If a fuel consumption rate of 7500 Btu/hp-hr is used for the uncontrolled unit, the heat rejection is increased by about 15 percent.

| Engine | Heat Exhausted to Atmosphere as Waste Heat from Uncontrolled Engine, Btu/yr | Increase Due to Control, Btu/yr |
|---|---|---|
| 500 hp | $20 \times 10^9$ | $3 \times 10^9$ |
| 13,500 hp | $535 \times 10^9$ | $81 \times 10^9$ |

## 7.6 OTHER ENVIRONMENTAL CONCERNS

### 7.6.1 Irreversible and Irretrievable Commitment of Resources

No irreversible or irretrievable commitment of resources is anticipated as a result of promulgating standards of performance for new stationary reciprocating engines other than that already discussed resulting from increased fuel consumption, and, possibly, use of catalysts.

### 7.6.2 Environmental Impact of Delayed Standards

The environmental impact of delaying a standard that is based on emission levels achieved by presently available control techniques (see Chapter 6) is essentially the same as the impact of no standard, since stationary reciprocating engines are presently unregulated for exhaust emissions except for local regulations for smoke and the Los Angeles area $NO_x$ regulation. The control techniques which require additional development (i.e., EGR on turbocharged engines, water induction, and combustion chamber modifications) can potentially achieve lower emission levels than those achieved by currently available techniques. However, the imposition of standards based on currently available technology does not preclude or interfere with the continued development of the emerging techniques. Moreover, the promulgation of a standard which prescribes an emission level that is based on current technology to take effect several years hence would not lead to wasteful commitment of resources (i.e., does not force engine manufacturers to expend large sums of money to develop one kind of engine or control system now and a completely unrelated kind for the future); the technology is at hand to enable manufacturers to meet a near-term standard, so that most R&D funds could be devoted to preparing for a lower, future standard. Therefore, no environmental gain is achieved by delaying promulgation of standards.

The impact of no standard relative to a standard based only on readily available control technology (i.e., a near-term standard) is discussed below. The additional improvement to the ambient air that can be obtained in the future through the application of more stringent standards based on emerging control technology is also considered.

### 7.6.3 Environmental Impact of No Standard

The estimated impact presented here for no standard is based on the same reasoning used for the estimated energy impact (see Section 7.4). Table 7-5(13) shows the impact of a $NO_x$ standard for 1 year's production of diesel, dual-fuel, and gas engines that would be affected by a standard of performance. Three possible levels of control (20, 40 and 60 percent) are considered based on the alternatives discussed in Chapter 6. This table shows that the impact of the standard could range from 0.06 to 0.19 percent of the total U.S. $NO_x$ emissions, depending on the level of control imposed.

TABLE 7-5. ESTIMATED ANNUAL IMPACT OF NO STANDARD ON U.S. $NO_x$ EMISSIONS[a]

| Fuel | No. Units Added Annually | Added hp x $10^3$ | Usage, hr/yr | Emission Rate,[b] g/hp-hr | Percentage $NO_x$ Reduction | Estimated $NO_x$ | | Decrease from Uncontrolled Emissions Metric tons/yr x $10^3$ | Decrease from Uncontrolled Emissions as a Percent of U.S. Total[c] |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | Metric tons/yr x $10^3$ | % U.S. Total | | |
| Diesel | 25 | 87 | 8,000 | 11 | 0 | 7.6 | 0.032 | -- | -- |
| | | | | 9 | 20 | 6.3 | 0.026 | 1.3 | 0.0055 |
| | | | | 7 | 40 | 4.9 | 0.021 | 2.7 | 0.0113 |
| | | | | 4 | 60 | 2.8 | 0.012 | 4.8 | 0.0202 |
| Dual Fuel | 12 | 30 | 8,000 | 8 | 0 | 1.9 | 0.008 | -- | -- |
| | | | | 6 | 20 | 1.4 | 0.006 | 0.5 | 0.0021 |
| | | | | 5 | 40 | 1.2 | 0.005 | 0.7 | 0.0029 |
| | | | | 3 | 60 | 0.7 | 0.003 | 1.2 | 0.0050 |
| Natural Gas | 400 | 524 | 8,000 | 15 | 0 | 63 | 0.265 | -- | -- |
| | | | | 12 | 20 | 50 | 0.210 | 13 | 0.0546 |
| | | | | 9 | 40 | 38 | 0.160 | 25 | 0.1050 |
| | | | | 6 | 60 | 25 | 0.105 | 38 | 0.1597 |
| Total | 437 | 641 | -- | -- | -- | -- | -- | 14.8 - 44[d] | 0.062 - 0.185[d] |

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[a] Decrease from 1 year's production of engines relative to 1974 total U.S. emissions

[b] Emission rates correspond to 20%, 40%, and 60% reduction in $NO_x$ from uncontrolled level, to the nearest 1 g/hp-hr. Uncontrolled levels are based on 5-year sales-weighted averages (see Section 4.3.4)

[c] Based on 23,800 x $10^3$ metric ton/yr from all sources (mobile and stationary) in 1974 (Reference 13)

[d] Range of decreased emissions corresponds to a 20% to 60% reduction from uncontrolled levels

These impacts are based on the assumption of constant production of diesel, dual-fuel, and natural gas engines.

Nevertheless, several projections conclude that total U.S. $NO_x$ emissions from stationary sources will increase at annual compounded rates of 3.9 to 6.4 percent(14,15). These projected increases are based on economic indicators which suggest a substantial industrial growth during the next 10 to 15 years. The growth in stationary source emissions is partially offset by declining $NO_x$ emissions from automobiles due to the application of controls. When both trends are considered, total nationwide emissions (stationary plus mobile) are projected to increase by a factor of 1.2 to 1.4 by 1985. Under these circumstances, and assuming only a 20-percent $NO_x$ reduction, the incremental $NO_x$ emissions impact in 1985 for all new controlled large-bore engines (all fuels beginning in 1979) would amount to about 0.27 percent of the 1985 U.S. $NO_x$ emissions from all sources. If a 60-percent reduction were required starting in 1985, the differential impact would rise to about 0.93 percent in 1990.

Industry spokesmen report that 3 to 4 percent of the present population of large-bore engines is being replaced each year. Assuming production remains constant and that a performance standard for emissions is in effect, then all of the present population of engines will be replaced with controlled engines in 30 years. This new population of controlled engines (assuming controls achieve a 60-percent $NO_x$ reduction) would effect a reduction in total U.S. $NO_x$ emissions of 5.6 percent based on present (1974) total emissions of $NO_x$, or a 2-percent reduction assuming $NO_x$ emissions increase 3.5 percent per year over this period.

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(15) Habegger, L. J., R. R. Cirillo and N. F. Sather. Priorities and Procedures for Development of Standards of Performance for New Stationary Sources of Atmospheric Emissions. Prepared by Argonne National Laboratory for EPA/OAQPS/ESED, May 1976, EPA-450/3-76-020.

# CHAPTER 8

# ECONOMIC IMPACT

This chapter is divided into four sections. Section 8.1 presents a general profile of the domestic stationary reciprocating engine industry. Emphasis is given to large-bore stationary engine manufacturers and their major end uses, since proposed New Source Performance Standards (see chapter 9) will affect primarily this industry. Section 8.2 is a detailed cost analysis of the alternative $NO_x$ control techniques summarized in Chapter 6. Section 8.3 is a discussion of potential costs to stationary engine manufacturers and users arising from other environmental regulations. The chapter concludes with a detailed analysis, in Section 8.4, of the economic impact that could arise from the use of the alternative emission control techniques to meet a performance standard.

## 8.1 INDUSTRY PROFILE

This section presents a business profile of the IC engine industry. It is divided into five subsections. In Section 8.1.1 manufacturers of domestic stationary engines are presented, and general production trends and applications are discussed. In Section 8.1.2 the discussion is focused on the large-bore engine manufacturers. A detailed analysis of large-bore engine markets is given in Section 8.1.3. Finally, a discussion of engine imports and exports is presented in Section 8.1.4.

### 8.1.1 Structure of the Industry

The stationary IC engine industry consists of 40 firms operating in four separate submarkets: (1) high-power, low-to medium-speed engines; (2) medium-power, medium to high-speed engines; (3) low-power, high-speed engines; and (4) very small one-cylinder gasoline engines (<20 hp). These manufacturers were listed by submarket in Table 3-1. Figures 3-1 and 3-2 depicted graphically the relationships between the first three groups on the basis of engine power. All 40 firms are listed in alphabetical order, with addresses, in Table 8-1. In addition, engine descriptions are given in Table 8-2(1) by manufacturer for all but some very small gasoline units.

Table 8-3(2-4) shows total U.S. production of IC engines by type of fuel (gasoline, diesel, and natural gas) for the years 1964 through 1975. These data are also presented in graph form, with a separate plot for each fuel, in Figures 8-1 to 8-3(5-6). These data represent production of all engines, irrespective of whether they were used in mobile or stationary applications. Production data of engines sold for stationary uses in 1973 have been estimated by direct contacts with manufacturers (see Table 3-2 and Section 8.1.3). These data have not been compiled previously for public dissemination by any government agency or trade association, nor is there any reliable basis for extrapolating the 1973 estimates to other years. The reader should note that these data are for shipments of engines produced by U.S. manufacturers and do not consider imports or exports. For a discussion of imports and exports see Section 8.1.4.

Several broad trends are indicated by these production data: (1) Gasoline engine production grew slowly up to 1974, with the bulk of the

## TABLE 8-1. U.S. MANUFACTURERS OF I.C. ENGINES FOR STATIONARY APPLICATIONS

Alco Power Incorporated
G.E.C Diesels Limited
100 Orchard Street
Auburn, New York 13021
Brand Name: Alco

Allis-Chalmers Corporation
Engine Division
151st and Halsted Street
Box 563
Harvey, Illinois 60426
Brand Name: Allis-Chalmers

Briggs and Stratton Corporation
3300 North 124th Street
Wauwatosa, Wisconsin 53201
Brand Name: Briggs & Stratton

J. I. Case Company
700 State Street
Racine, Wisconsin 53409
Brand Name: Case

Caterpillar Tractor Company
100 N.E. Adams Street
Peoria, Illinois 61602
Brand Name: Caterpillar

Chevrolet-Oldsmobile Division
General Motors Corporation
General Motors Building
Detroit, Michigan, 48202
Brand Name: Chevrolet-Oldsmobile

Chrysler Corporation
Marine and Industrial Products Plant
Marysville, Michigan
Brand Name: Chrysler

Clinton Engines Corporation
Maquoketa, Iowa 52060
Brand Name: Clinton

Colt Industries
Power Systems Division
701 Lawton Avenue
Beloit, Wisconsin 53511
Brand Name: Fairbanks Morse

Cooper Energy Services
North Sandusky Street
Mt. Vernon, Ohio 43050
Brand Name: Cooper-Bessemer

Ajax Division
Cooper Energy Services
19 North Center
Corry, Pennsylvania 16407
Brand Name: Ajax

Cummins Engine Company
1000 Fifth Street
Columbus, Indiana 47201
Brand Name: Cummins

Detroit-Diesel Allison Division
General Motors Corporation
13400 West Outer Drive
Detroit, Michigan 4828
Brand Name: Detroit-Diesel

Dresser-Clark Division
Dresser Industries Incorporated
600 Lincoln Avenue
Olean, New York 14760
Brand Name: Clark

ElectroMotive Division
General Motors Corporation

9301 55th Street
La Grange, Illinois 60520
Brand Name: Electro-Motive

Enterprise Engine and
Compressor Division
DeLaval Turbine Incorporated
550 85th Avenue
Oakland, California 94621
Brand Name: Enterprise

Ford Motor Company
Industrial Engines Operation
Dearborn, Michigan 48121
Brand Name: Ford

TABLE 8-1. Continued

General Electric Company
Diesel Power Products
Transportation Business Division
2901 East Lake Road
Erie, Pennsylvania 16531
Brand Name: General Electric

Homelite Division of Textron
70 Riverdale Avenue
Port Chester, New York 10573
Brand Name: Homelite

Ingersoll-Rand Company
Engine Process Compressor Division
100 West Chemung Street
Painted Post, New York 14870
Brand Name: Ingersoll-Rand

International Harvester Company
10400 West North Avenue
Melrose Park, Illinois 60160
Brand Name: International

Jacobsen Manufacturing Company
1721 Packard Avenue
Racine, Wisconsin 53403
Brand Name: Jacobsen

John Deere OEM Sales
Moline, Illinois
Brand Name: John Deere

Kohler Company
Kohler, Wisconsin 53044
Brand Name: Kohler

McCulloch Corporation
6101 Century Blvd.
Los Angeles, California 90045
Brand Name: McCulloch

Murphy Diesel Company
5317 W. Burnham Street
Milwaukee, Wisconsin 53219
Brand Name: Murphy Diesel

O&R Engines Incorporated
3340 Emery Street
Los Angeles, California 90023
Brand Name: O&R

Outboard Marine Corporation
Gale Products Division
100 Sea Horse Drive
Waukegan, Illinois 60085
Brand Name: Lawnboy

Sterling Engine Company, Inc.
3600 NW North River Drive
Miami, Florida 33142
Brand Name: Sterling

Stewart and Stevenson Services, Inc.
4516 Harrisburg Boulevard
Houston, Texas 77011
Brand Name: Stewart & Stevenson (Modified Detroit-Diesel Engines)

Tecumseh Products Company
Ottawa and Patterson Streets
Tecumseh, Michigan 49286
Brand Name: Tecumseh

Teledyne Continental Motors
Industrial Products Division
205 Market Street
Muskegon, Michigan 49443
Brand Name: Continental

Teledyne Wisconsin Motors
1910 South 53rd Street
Milwaukee, Wisconsin 53246
Brand Name: Wisconsin, Hatz

Waukesha Engine Division
Dresser Industries Inc.
100 West St. Paul Street
Waukesha, Wisconsin 53186
Brand Name: Waukesha

White Construction
Equipment Division
White Motor Corporation
Hopkins, Minnesota 55343
Brand Name: Minneapolis Moline

## TABLE 8-1. Concluded

White Engines Incorporated
Hercules Engine Division
101 11th street S.E.
Canton, Ohio 44702
Brand Name: Hercules

Superior Division
Cooper Energy Services
1401 Sheridan Avenue
Springfield, Ohio 45505
Brand Name: White Superior

Wills Industries
Dieselec Division
Harvey, Louisiana 70058
Brand Name: Dieselec

Witte Engine Corporation
555 East 56 Highway
Olathe, Kansas 66061
Brand Name: Witte

## TABLE 8-2. STATIONARY ENGINES MANUFACTURED IN THE UNITED STATES (1)

| Manufacturer | Engine Series | Fuel[a] | Power[b], hp | bmep, psi | Speed, rpm | Strokes per Cycle | Air Charging | | | | Fuel Charging | | | Comments |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | Naturally Aspirated | Atmospheric Blower | Turbocharged | Intercooled | Direct Injection | Indirect Injection | Carbureted | |
| Alco | 251 | D | 1000-4400 | 215-290 | 400-1200 | 4 | | | X | X | X | | | |
| Allis-Chalmers | | D | 29-850 | 100-170 | 1200-2600 | 4 | X | | X | X | X | | | |
| J.I. Case | | D | 52-123 | 105-142 | 1200-2200 | 4 | X | | X | | X | | | |
| | | G | 52-123 | 110 | 1200-2400 | 4 | X | | | | | | | |
| Caterpillar | | D | 170-1000 | - | 1200 | 4 | X | | X | X | | X | | |
| | | D | 85-890 | 164 | 1800-2000 | 4 | X | | X | X | | X | | |
| | | D | 125-175 | 90 | 2400 | 4 | X | | X | | X | | | |
| | | G | 200-930 | - | 1200 | 4 | X | | X | X | | | | |
| | | G | 125-350 | - | 1800 | 4 | X | | X | X | | | | |
| Chevrolet Oldsmobile | 153<br>250<br>292<br>427<br>455 | Gl<br>G | 50-215 | 72-103 | 3200-3600 | 4 | X | | | | | | X | |
| Chrysler | | Gl | 43-175 | 95-120 | 2000-4000 | 4 | X | | | | | | X | |
| Colt | PC2 | DF | 5700-9400 | 228 | 514 | 4 | | | X | X | X | | | Liscense from S.E.M.T.[c] opposed piston |
| | 3800$\frac{1}{8}$ | DF,G | 850-4200 | 190-220 | 720-900 | 2 | | X | X | | X | | | |

[a] D - Diesel, DF - Dual Fuel, G - Gas (spark-ignited), Gl - Gasoline (spark-ignited)

[b] Continuous rating

[c] Societe´ des Etude Mecanique et Thermique, France

## TABLE 8-2. (Continued)

| Manufacturer | Engine Series | Fuel[a] | Power,[b] hp | bmep, psi | Speed, rpm | Strokes per Cycle | Air Charging |  |  |  | Fuel Charging |  |  | Comments |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  |  |  |  | Naturally Aspirated | Atmospheric Blower | Turbocharged | Intercooled | Direct Injection | Indirect Injection | Carbureted |  |
| Cooper-Bessemer | LSV, KSV | DF,G | 4300-7800 | 140-190 | 400-600 | 4 |  |  | X | X | X |  |  |  |
|  |  | G | 900-13,500 | 80-128 | 275-330 | 2 |  | X | X |  |  |  |  | Integral engine compressor |
| Cooper-Ajax |  | G | 15-300 | 50-59 | 200-900 | 2 |  | X |  |  |  |  |  | Gas injection (direct) |
| Cummins |  | D | 123-1200 | 140 | 1800-3000 | 4 | X |  | X | X | X |  |  |  |
| Detroit Diesel | 53 | D | 64-140 | 73 | 2200-2400 | 2 |  | X |  |  | X |  |  |  |
|  | 71 | D | 48-800 | 132 | 1800-2100 | 2 |  | X | X | X | X |  |  |  |
|  | 92 | D | 225-800 | - | 1800-2100 | 2 |  | X | X |  | X |  |  |  |
|  | 149 | D | 675-1080 | 100 | 1800-1900 | 2 |  | X | X | X | X |  |  |  |
|  |  | D | 56-116 | 93 | 2400-2500 | 4 | X |  |  |  | X |  |  |  |
| Dresser-Clark |  | G | 1000<br>11,000 | - | 300-330 | 2 |  | X | X |  |  |  |  | Integral engine compressor |
| Electro-Motive | 645 | D | 800-3600 | 82-130 | 900 | 2 |  | X | X | X | X |  |  |  |

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[a] D - Diesel, DF - Dual Fuel, G - Gas (spark-ignited), Gl - Gasoline (spark-ignited)

[b] Continuous rating

## TABLE 8-2. (Continued)

| Manufactuer | Engine Series | Fuel[a] | Power,[b] hp | bmep, psi | Speed, rpm | Strokes per Cycle | Air Charging |  |  |  | Fuel Charging |  |  | Comments |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  |  |  |  | Naturally Aspirated | Atmospheric Blower | Turbocharged | Intercooled | Direct Injection | Indirect Injection | Carbureted |  |
| John Deere | 300, 400, 500 | D | 44-180 | 80-130 | 1500-2200 | 4 | X |  | X | X | X |  |  |  |
| Enterprise (De Laval) | GVB | G,DF | 2900-5000 | 320 | 450 | 4 | X |  | X | X | X |  |  |  |
|  | R,RV | DF | 1600-13,500 | 100-250 | 450 | 4 | X |  | X | X | X |  |  |  |
|  | HVC | G | 2000-5500 | 300 | 630 | 4 | X |  |  |  |  |  |  | Engine for compressor module |
| Ford |  | G1,G | 38-200 | - | 2800 | 4 | X |  |  |  |  |  | X |  |
|  | DF | D | 33-124 | 110 | 3200-2500 | 4 | X |  | X |  | X |  |  |  |
| Ingersoll-Rand | PKVGR, PKVSR | G | 1080-3500 | 80-129 | 360 | 4 | X |  | X |  |  |  |  |  |
|  |  | G | 1000-5500 | 80-156 | 330-550 | 4 | X |  | X |  |  |  |  | Integral engine compression |
| International Harvester | D | D | 57-325 | 80-150 | 2400 | 4 | X |  | X | X | X |  |  |  |
|  | C,GT | G1,G | 16-197 | 109 | 1800-3000 | 4 | X |  |  |  |  |  | X |  |
| Kohler |  | G1, G | 3.8-28 | - |  |  |  |  |  |  |  |  |  | Electric Generator Sets |
|  |  | D | 7.6-28 | - |  |  |  |  |  |  |  |  |  | Electric Generator Sets |

[a] D - Diesel, DF - Dual Fuel, G - Gas (spark-ignited) G1 - Gasoline (spark-ignited)

[b] Continuous rating

TABLE 8-2. (Continued)

| Manufacturer | Engine Series | Fuel[a] | Power[b], hp | bmep, psi | Speed, rpm | Strokes per Cycle | Air Charging |  |  |  | Fuel Charging |  |  | Comments |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  |  |  |  | Naturally Aspirated | Atmospheric Blower | Turbocharged | Intercooled | Direct Injection | Indirect Injection | Carbureted |  |
| Minneapolis Moline | HD | G | 92-174 | 95.7 | 1800 | 4 | X |  |  |  |  |  |  |  |
|  | DHD | D | 105 | 92 | 1800 | 4 | X |  |  |  | X |  |  |  |
| Murphy | A/D | D | 125-524 | 101-170 | 1800 | 4 | X |  | X |  | X |  |  |  |
|  | MP | D | 112-335 | 100-148 | 1200-1400 | 4 | X |  | X |  | X |  |  |  |
| Onan | DJ | D | 7.9-29.8 |  | 2400 | 4 | X |  |  |  | X |  |  |  |
|  | BF,CCK, NH | Gl | 16-25 |  | 2700-3900 | 4 | X |  |  |  |  |  | X |  |
| Sterling | IH,IA, ID | D | 16-152 | 75 | 1200-1800 | 4 | X |  |  |  | X |  |  |  |
| Stewart & Stevenson | SG | G | 30-500 | - | 1200-1800 | 2 |  | X |  |  |  |  |  | Modified Detroit Diesel Series 71 |
|  |  | DF | 70-1000 | - | 1800 | 2 |  | X | X | X | X |  |  | Dual fuel conversion of Detroit diesel engines |
| Teledyne Continental |  | Gl | 50-80 | 60 | 2000 | 4 | X |  |  |  |  |  | X |  |

[a] D - Diesel, DF - Dual Fuel, G - Gas (spark-ignited), Gl - Gasoline (spark-ignited)

[b] Continuous rating

## TABLE 8-2. (Concluded)

| Manufacturer | Engine Series | Fuel[a], | Power[b], hp | bmep, psi | Speed, rpm | Strokes per Cycle | Air Charging |  |  |  | Fuel Charging |  |  | Comments |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  |  |  |  | Naturally Aspirated | Atmospheric Blower | Turbocharged | Intercooled | Direct Injection | Indirect Injection | Carbureted |  |
| Teledyne Wisconsin |  | D | 3.5-80 | - | 3000 | 4 | X |  |  |  | X |  |  | Air cooled |
|  |  | Gl, G | 4-66 | - | 2400-3600 | 4 | X |  |  |  |  |  | X | Air cooled |
| Waukesha |  | D | 52-1403 | 93-160 | 1200-1800 | 4 | X |  | X | X | X |  |  |  |
|  |  | G<br>Gl | 42-1540 | 93-160 | 1000-1800 | 4 | X |  | X | X |  |  | X |  |
| White Engines (Hercules) |  | Gl, G | 58-132 | - | 2400-2800 | 4 | X |  |  |  |  |  | X |  |
|  |  | D | 34-120 | 83 | 2400 | 4 | X |  |  |  | X |  |  |  |
| Superior Division Cooper Energy Services | G510-GT825 | G | 400-2650 | 52-147 | 900-1000 | 4 | X |  | X |  |  |  |  |  |
|  | 40 | DF | 220-2400 | 48-150 | 1000 | 4 | X |  | X |  | X |  |  |  |
|  | PTD | D | 400-2400 | 48-150 | 1000 | 4 | X |  | X |  | X |  |  |  |
| Wills Ind. (Dieselec) | A1D, A2D | D | 11-28 | - | 3000 | 4 | X |  |  |  | X |  |  | Air cooled |
| Witte |  | D | 9-27 | 88 | 800-1800 | 4 | X |  |  |  | X |  |  |  |
| General Electric | 7FDS | D | 1500-3500 | 282 | 1000-1200 | 4 | X |  |  |  | X |  |  |  |

[a] D - Diesel, DF - Dual Fuel, G - Gas (spark-ignited), Gl - gasoline (spark-ignited)

[b] Continuous rating

[c] Worthington ceased production of stationary I.C. engines during the preparation of this document

## TABLE 8-3. U.S. PRODUCTION OF IC ENGINES BY TYPE OF FUEL AND END APPLICATION 1964-1975[a]

| Fuel & Application | 64 | 65 | 66 | 67 | 68 | 69 | 70 | 71 | 72 | 73 | 74 | 75 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **Gasoline** | | | | | | | | | | | | |
| Marine & Garden | 5,275,318 | 6,249,609 | 7,008,314 | 7,640,224 | 8,321,317 | 8,824,557 | 8,675,614 | 8,367,304 | 9,382,361 | 11,314,740 | 11,958,996 | 8,933,338 |
| Agriculture[b] | 434,175 | 417,507 | 509,543 | 202,167 | 193,380 | 187,437 | 169,977 | 105,754 | 223,887 | 303,372 | 414,025 | 324,912 |
| Generator Sets | 59,190 | 67,769 | 76,678 | 67,930 | 67,798 | 90,760 | 86,364 | 104,142 | 146,270 | 165,183 | 176,014 | |
| General Industrial | 949,007 | 1,087,760 | 1,173,939 | 1,070,887 | 1,134,638 | 1,249,185 | 1,073,564 | 1,058,104 | 1,115,420[c] | 1,222,256 | 1,078,943 | 1,134,932 |
| Construction[d] | 66,052 | 85,076 | 132,214 | 121,225 | 104,638 | 175,605 | 152,178 | 116,074 | 140,992 | 177,544 | 193,608 | 129,803 |
| Total | 6,733,722 | 7,907,723 | 8,900,488 | 9,102,397 | 9,821,771 | 10,527,544 | 9,557,607 | 9,751,370 | 11,008,930 | 13,183,095 | 13,821,506 | 10,522,985 |
| **Diesel** | | | | | | | | | | | | |
| Marine | 47,111 | 56,305 | 7,213 | 9,764 | 9,762 | 8,604 | 6,745 | 5,771 | 7,686 | 9,083 | 8,940 | 8,230 |
| Construction[d] | | | 98,865 | 90,338 | 92,932 | 104,284 | 91,043 | 85,071 | 103,430 | 130,395 | 132,256 | 101,685 |
| Agriculture[e] | 118,064 | 103,325 | 94,503 | 102,459 | 96,460 | 80,264 | 65,641 | 75,017 | 91,104 | 112,068 | 128,223 | 132,643 |
| Generator Sets | | | 12,746 | 5,564 | 6,070 | 8,535 | 10,201 | 8,400 | 9,661 | 13,327 | 15,212 | |
| Locomotive[f] | 9,548 | 13,209 | | | | | | | | | | 92,558 |
| General Industrial | 63,320 | 73,880 | 41,156 | 44,327 | 46,645 | 52,045 | 52,218 | 45,145 | 47,393 | 44,676 | 67,798 | |
| Total | 237,043 | 246,719 | 254,489 | 252,452 | 251,859 | 253,732 | 225,853 | 219,344 | 259,274 | 309,549 | 352,429 | 335,116 |
| **Gas** | | | | | | | | | | | | |
| Agriculture | 5,780 | 5,654 | 11,460 | 6,873 | 3,947 | 3,257 | approx. 3,000 | 2,341 | 1,849 | 2,589 | 7,829 | |
| Construction | | | | | | | | 307 | 4,965 | | 201 | 5,124 |
| Generator Sets | 7,700 | 8,256 | 5,539 | 4,799 | 3,547 | 2,696 | approx. 2,300 | | | | | |
| General Industrial[g] | 911 | 839 | 1,135 | 1,260 | 1,028 | 1,002 | 1,821 | | | | | |
| Total | 14,391 | 14,759 | 18,134 | 12,932 | 8,522 | 6,953 | 7,150 | 5,574 | 11,145 | 9,419 | 8,030 | 5,124 |

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[a] Except automotive, outboard, and aircraft engines(2,3).

[b] Over 75 percent of gasoline agricultural engines are for motive power such as tractors, combines, etc.(4).

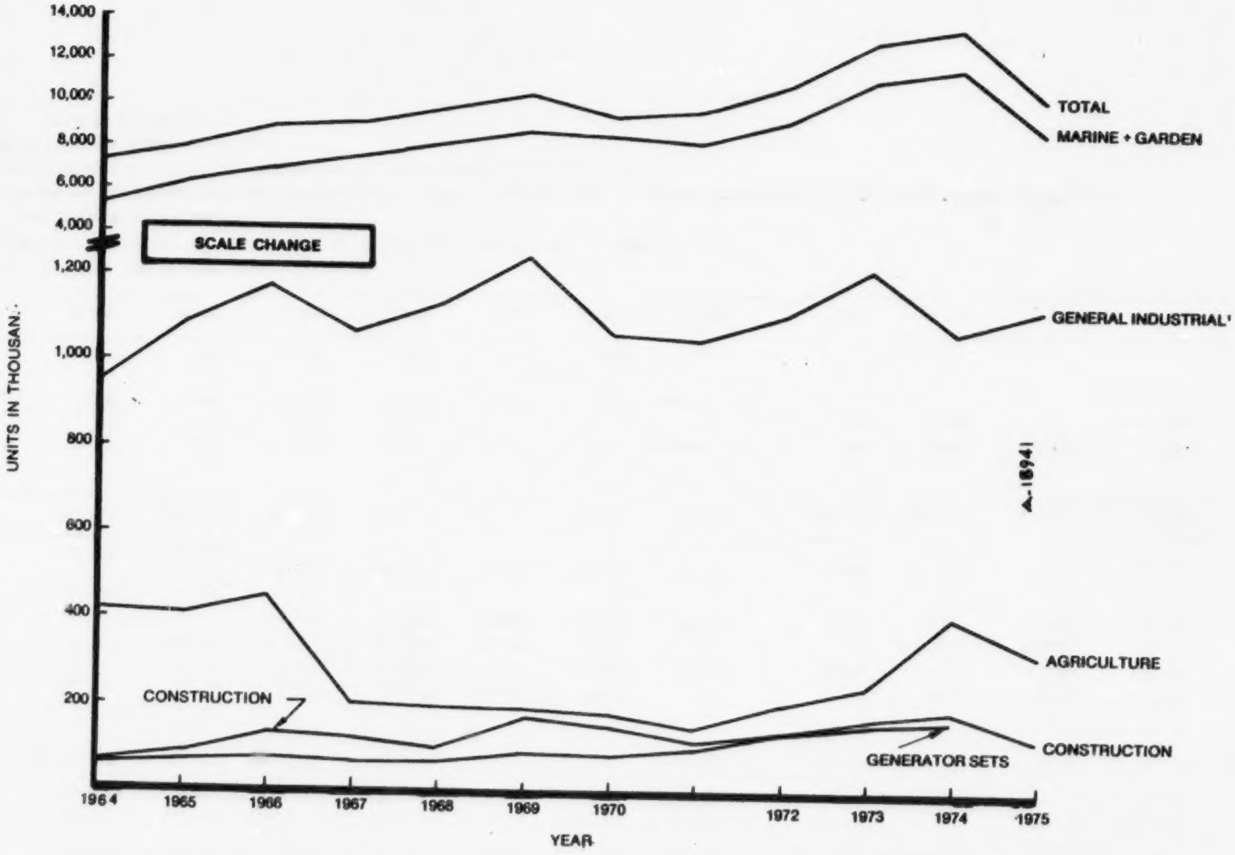
[c] 1972 figure includes railroad motive power.

[d] Most construction engines are motive power applications such as loaders, scrapers, etc. Stationary applications include pumps, blowers, drills, etc.

[e] More than 50 percent of agricultural diesel engines are for nonstationary applications(4).

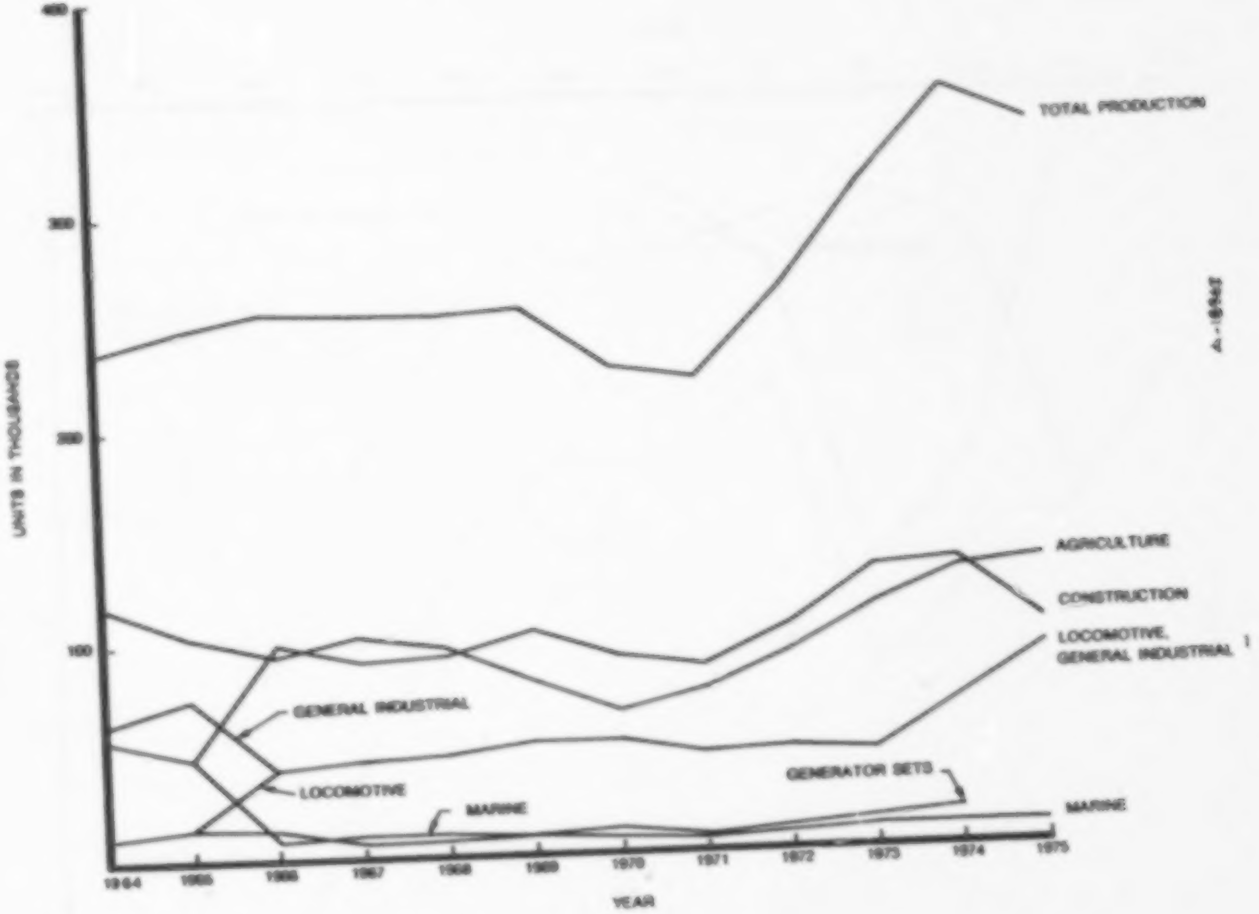
[f] 1000-2000 locomotive engines produced each year.

[g] General industrial added to agriculture.



¹INCLUDES GENERATOR SETS FOR 1975 ONLY.

Figure 8-1. Annual production of gasoline engines (excluding automotive, outboard, and aircraft engines). (5,6)



[1] Includes generator sets for 1975 only.

Figure 8-2. Annual production of diesel engines (excluding automotive, outboard, and aircraft engines)(5,6).

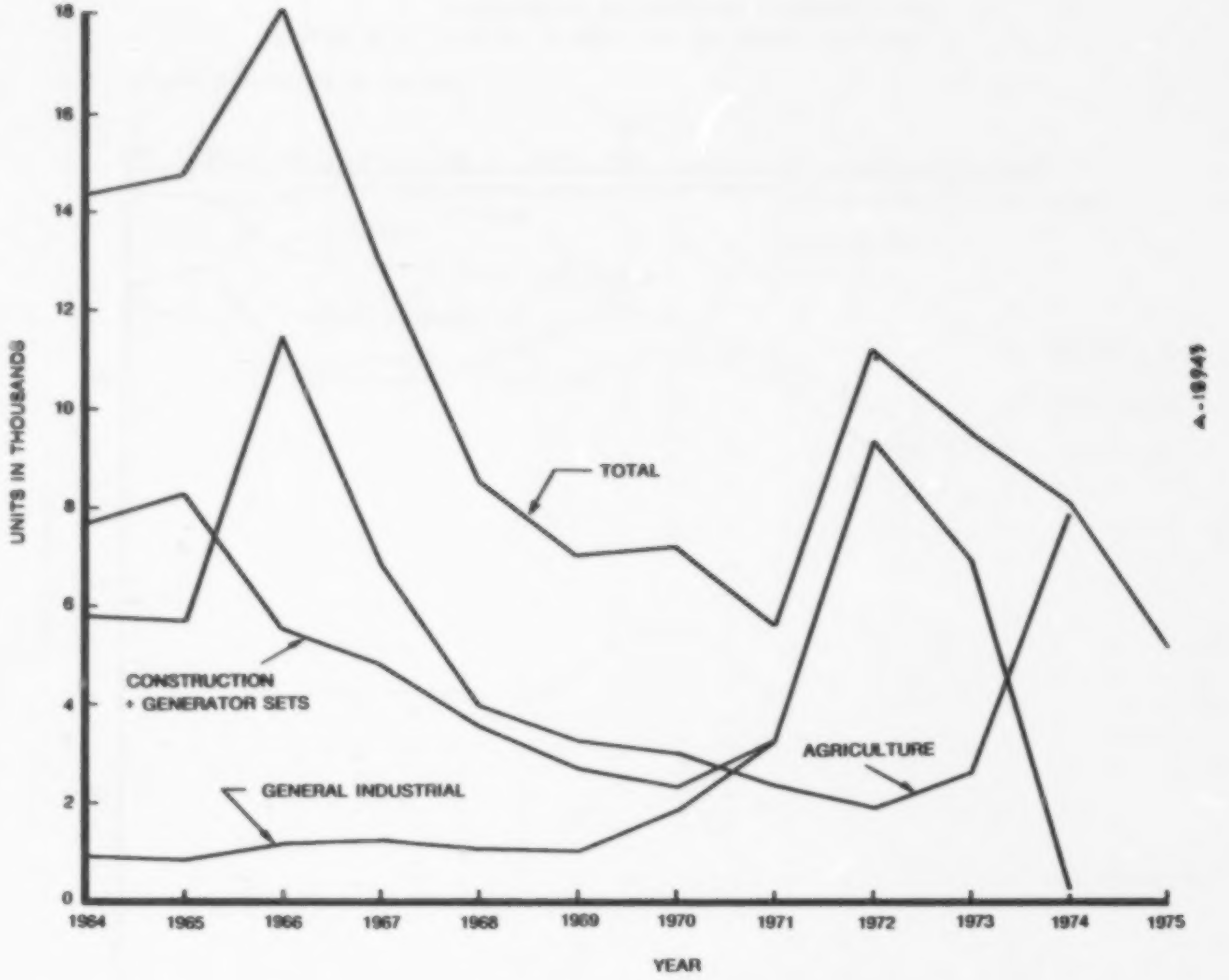


Figure 8-3. Annual production of gas engines (excluding automotive, outboard, and aircraft engines)(5,6).

increased production occurring in marine and garden applications (small engines, typically much less than 15 hp). The 1975 production, however, declined coinciding with a general decline in the economy. (2) The diesel fuel category experienced a much more rapid growth rate up to 1974 but also declined in 1975. Note that the 1975 sales were nearly evenly distributed among construction, agriculture, generator sets, and general industrial applications. (3) Natural gas engine production has been declining rapidly since 1966. According to industry representatives this sharp drop in sales of gas-fueled engines reflects the decline in usage of IC engines for irrigation pumps and pipeline compressors(7). (Ninety percent of the installed reciprocating engine horsepower in pipeline applications is spark-ignited; slightly over two-thirds of the remainder is dual-fueled)(8). This trend is the result of the unavailability of natural gas for irrigation and the end of the nation's expansion of interstate natural gas pipelines.

### 8.1.2 Large-Bore Engine Manufacturers

The large-bore stationary engine industry consists of nine firms manufacturing engines in the range of 400 to 13,500 hp and 300 to 1200 rpm. In general, the engines produced by this industry are designed for heavy duty operation at high power outputs and for long lifetime (hence the low rpm ratings). The primary applications of these engines are for (1) oil and gas production, (2) oil and gas transmission, (3) electric generation and (4) standby service (nuclear power plants, standby generators, and floodpumps). These applications are discussed in more detail in Section 8.1.3.

As shown in Table 8-4 the corporations manufacturing high-power stationary engines are all highly diversified, and many are among the 500

## TABLE 8-4. MANUFACTURERS OF HIGH-POWER ENGINES

| Manufacturer | Parent Corporation | Rank in "Fortune's 500"[a] |
|---|---|---|
| Alco Power Inc. | G.E.C. Limited | NA[b] |
| Caterpillar | Caterpillar Tractor Co. | 36 |
| Cooper-Bessemer<br>Superior<br>Ajax | Cooper Energy Services | 340 |
| Dresser-Clark Division<br>Waukesha Division | Dresser Industries | 101 |
| Enterprise Engine Division of DeLaval Turbine Co. | TransAmerica | NM[c] |
| ElectroMotive Division | General Motors Corporation | 2 |
| Fairbanks Morse Engine Division | Colt Industries | 178 |
| General Electric | General Electric | 9 |
| Ingersoll-Rand | Ingersoll-Rand | 117 |

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[a] Based on 1976 sales data (9).

[b] NA = Not applicable, Alco Power Inc., Division White Industrial Power was purchased by G.E.C. Ltd of Great Britain, a conglomerate of engine manufacturers and related firms.

[c] NM = Not meaningful since TransAmerica is primarily a financial corporation, rather than an industrial firm.

largest industrial firms in the U.S. Seven of the nine manufacturers of large-bore engines either market gas turbines or belong to a parent company which markets turbines through another division. Alco, General Electric, and ElectroMotive manufacture engines primarily for the railroad industry. General Electric has only recently begun marketing engines for stationary applications, and primarily exports for drilling and auxiliary power applications. Some of the firms also manufacture large-bore engines for marine propulsion and mobile applications (e.g., off-highway construction vehicles). As shown in Table 8-5(10), all of these organizations are part of diversified parent corporations for which sales of stationary reciprocating engines account for less than 15 percent of total revenues.

Since 1973, three firms -- Chicago Pneumatic, Nordberg, and Worthington -- have ceased production of stationary reciprocating engines. Conversations with industry spokesmen indicate that the industry has stabilized, and no additional dropouts are anticipated.

Manufacturing plants employ from 600 to 10,000 people each. In the larger plants that produce engines for other than stationary applications, approximately 600 to 2000 employees per plant can be allocated to manufacturing stationary engines. Three of the nine plants which produce large engines employ less than 1000 people.

Sales of high-power stationary engines are comparatively unconcentrated; the largest manufacturer has approximately five-times the dollar volume of the smallest. Unit sales range from about 20 to 200 units per year for stationary applications depending on the manufacturer.

## TABLE 8-5. PARENT CORPORATION DATA

| Parent Corporation | Total Sales[a] ($ millions) | Rank in 1976 Fortune 500 |
|---|---|---|
| Dresser Industries (Dresser-Clark, Waukesha) | $ 2,232 | 101 |
| Ingersoll-Rand | 1,922 | 117 |
| Colt Industries (Fairbanks Morse) | 1,267 | 178 |
| Cooper Industries (Cooper, Superior, Ajax) | 554 | 340 |
| General Motors Corp. (ElectroMotive Div.) | 47,181 | 2 |
| Caterpillar Tractor Co. | 5,042 | 36 |
| TransAmerica (DeLaval) | NM[b] | NM[b] |
| General Electric | 15,697 | 9 |
| All Firms | $554-47,181 | 2-340 |

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[a] Total sales are for 1976. (Reference 10)

[b] NM = Not meaningful because TransAmerica is primarily a financial corporation, instead of an industrial corporation.

In general, firms which make smaller engines have a higher unit volume. For example, the total engine sales of Caterpillar or Waukesha are many times that of any other manufacturer. Furthermore, 90 percent of ElectroMotive's sales are for locomotive power, and their stationary sales are comparable in number and dollar volume to their competitors.

Industry representatives report that purchased materials account for approximately 60 percent of the cost of an engine. Most manufacturers purchase large forgings in finished or semifinished form, while some, for example, DeLaval and Superior Division/Cooper Energy Services, have their own facilities and make their own large castings. Turbochargers, aftercoolers, radiators, fuel systems, governors, oil pumps and other auxiliary systems are generally purchased from outside vendors.

Sales of high-power, low-speed engines are usually made through competitive bids that include a fuel consumption guarantee and the cost of providing a complete engine installation. This most commonly includes foundations, cooling systems, and other components such as generators and compressors. Often the cost of the basic engine is as little as 40 percent of the total(11). With the exception of standby service, most applications of large-bore engines are high usage (>2000 hr/yr). Thus fuel and maintenance costs, rather than initial price, are the most significant cost items for the purchaser of an engine. Nevertheless, manufacturing industry spokesmen indicated that engine prices have risen rapidly due to increases in raw material and labor costs.

It is anticipated that the financial impacts on the manufacturers of engines which must meet a standard of performance will be primarily in the form of research and development costs rather than costs to modify or retool the manufacturing process (see Section 8.4). Unlike standards of performance for new sources that are applied to typical manufacturing processes, the ones envisioned for IC engines are directed at the product of engine manufacturers rather than at the plants. Hence, controlling engine emissions will primarily involve modifications within the product. Although attempts to reduce engine emissions may require additions or alterations of one or two machines on an assembly line, the costs of retooling that could be required for EGR systems, new fuel pumps and injectors, new head designs, etc., can be considerable. Manufacturers have advised us that they usually do not proceed with the acquisition of the new tools or machines until the engine modifications habe been thoroughly tested on laboratory engines and their durability verified(12,13).

Research and development costs are normally deducted from income during the year incurred and not amortized over several years production of a specific engine; however, a company's engines are usually priced to recover these overhead expenses. Spokesmen for large-bore engine manufacturers indicate that the R&D programs typically span 3 to 5 years but may require as long as 10 years, with about 1 year of endurance testing required before any final design can be incorporated in a production model(14). The final development work is often carried out in cooperation with a well-known customer who receives the engines on an economically attractive basis. The level of development work that can be undertaken, however, will vary depending on the manufacturer and the applications the manufacturer serves.

### 8.1.3 Stationary Markets for Large-Bore Reciprocating Engines

The principal applications of large-bore stationary reciprocating engines are listed below:

| Application | Type of Engine |
|---|---|
| Oil and Gas Production | Gas and Diesel |
| Oil and Gas Transmission | Gas |
| Electric Generation (baseload) | Dual-Fuel and Diesel |
| Standby Service | Diesel |

Oil and gas production applications consist of engine-driven compressors for field gathering, pressurization, storage, and distribution, and oil and gas transmission applications for engine-driven compressors installed on natural gas pipelines. Dual-fuel and diesel engines are purchased primarily by municipal electric utilities for baseload (continuous) electric generation. Standby service includes emergency power for buildings and hospitals as well as nuclear generating stations and flood control pumps. Miscellaneous applications of large-bore engines include auxiliary engines on construction equipment (e.g., hoists and dredges) material handling, and pumping and electric generation at sewage treatment plants.

Sales of large-bore engines have increased slightly from 1- to 1-1/4-million horsepower per year over the past 5 years (1972 to 1976). Approximately 20- to 30-percent of this horsepower is exported, primarily as electrical generator units and engine/compressors for oil and gas production. Exports have increased during the past 5 years, and account for more than 50 percent of total sales for one domestic manufacturer. In general, spokesmen for manufacturers have indicated that rapid increases in sales for export and standby applications have offset depressed sales for natural gas pipelines and municipal electric utilities.

The following discussion of engine application is subdivided by fuel type since this classification provides a clear distinction of end uses. Aggregated sales information are presented below for large-bore engines by fuel type. These data were compiled from responses of engine manufacturers to a Section 114 Request for Information mailed in June of 1976 by EPA.(15) EPA has decided not to release data for individual manufacturers since manufacturers consider this information proprietary.

### 8.1.3.1 Diesel Engine Markets

The primary high usage (large emissions impact), domestic application of large-bore diesel engines during the past 5 years has been for oil and gas exploration and production. These and other applications are illustrated in Figure 8-4(16-21). As this figure shows, the market for prime (continuous) electric generation and other industrial applications all but disappeared after the 1973 oil embargo, but was quickly replaced by sales of standby electric units for building services, utilities, and nuclear power stations. The rapid growth in the oil and gas production market occurred because diesel units are being used on oil drilling rigs of various sizes. Sales of engines to export applications have also grown steadily since 1972, and are now a major segment of the entire sales market.

Some degree of overlap between large-bore diesels exists for petroleum exploration applications. Smaller (250 to 1000 hp) medium-bore designs (e.g., Detroit Diesel, Cummins, and Caterpillar) are used on portable drilling rigs to drill or service 2500- to 5000-feet wells. These rigs are trailer-mounted or helicopter-transported; therefore, small lightweight (approximately 4000-lb) engines are favored. In addition,

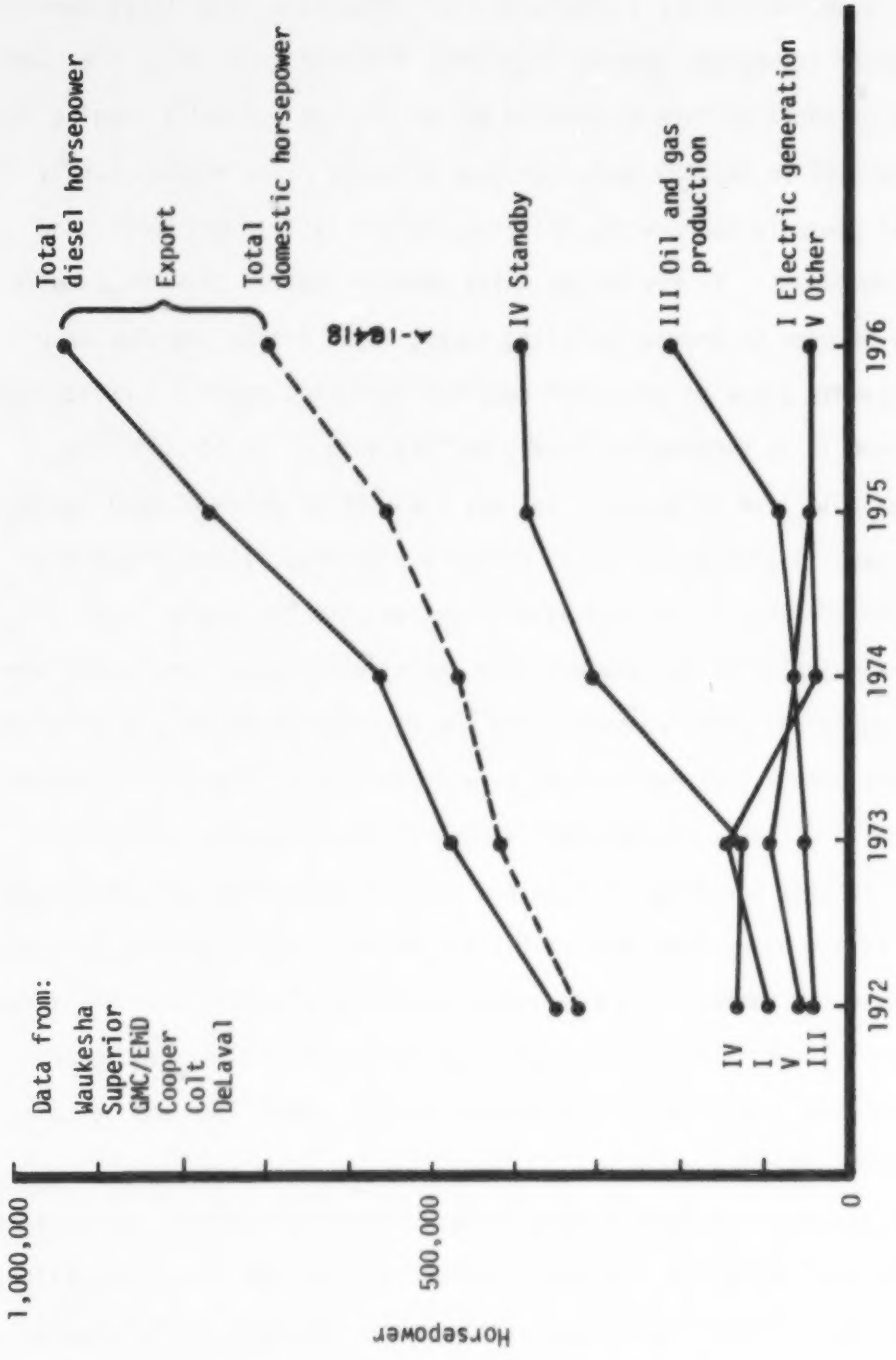

Figure 8-4. Sales of large-bore (>350 CID/cyl) diesel horsepower (from References 16 to 21).

multiple units are preferred to ensure some backup power in the event one engine is down, ruling out a single unit of comparable total horsepower.

Larger horsepower engines (Waukesha, ElectroMotive, etc.) are used in groups of three to five to provide 800 to 3000 hp for wells ranging in depth from 5000 to 25,0000 feet. On most of these rigs, engines supply mechanical power to operate the drilling (rotary table), mud pumps, and hoisting equipment. In the larger units several engines from one manufacturer may be used to provide drilling power, while another manufacturer's engines operate pumps or generator sets for auxiliary power. A relatively new approach is to generate AC power, rectify some of it for drilling power (variable load DC motors), and use the rest to drive AC auxiliaries. This approach is used primarily on offshore platforms, although there is interest in applying it to land-based sites despite its higher cost.

The majority of the engines used for electric power generation are owned by municipal power companies and are used for baseload generation in areas where demand does not justify construction of a large steam generating plant(22). Since the mid-1960's most of these engines have been dual fuel to take advantage of both low natural gas prices and compression ignition fuel economy (see Section 8.1.3.2 below). These engines are concentrated in the Northern Prairie states where municipally-owned utilities are common. In addition, engines are used for remote electrical power generation at mining sites in the Western states, where inexpensive commercial power is not available.

At present, the fuel economy of reciprocating IC engines gives them a distinct cost advantage over gas turbines for baseload power generation, while the lower initial cost makes turbines more competitive for peaking and standby applications. The relationships between cost and annual utilization are shown for these two power sources, plus fossil steam and nuclear steam, in Figure 8-5(23). Note that the figure is based on 1968 information; however, the relative comparison of these prime movers is still valid. These results do not include the cost impact of standards of performance for gas turbines. This impact is discussed in Section 8.4.1.3

As of January 1, 1974, six new IC engine generators were scheduled to be added to commercial power supplies -- five (total 32,840 kW) in 1974 and one (4415 kW) in 1975(24). This compares with six engines (30,930 kW) added in 1973. The Federal Power Commission reports that as of April 1, 1977, 565,000 kW (39 units) of diesel and dual fuel generating capacity were scheduled to be installed in the period 1977 to 1986(25). Since most of the IC engines used for electric power generation are owned by municipal utilities, which are generally smaller than the investor-owned utilities, it is possible that uncertainties in fuel availability and current high interest rates are preventing these smaller, municipally owned systems from raising the capital necessary to expand their systems. A spokesman for one manufacturer, however, stated that sales have picked up as the demand for additional power has reached critical level. In addition, another source believes that an increasing number of engines will be used for onsite power generation by municipalities and large industrial electricity users(26).

Large diesels are also used in nuclear powerplants, since these facilities are required to have emergency power available to flood the reactor core with water in the event of a reactor failure. Industry representatives (both manufacturers and users) have indicated that the high-power diesel engines have no effective competition for this market(27). Due to the quick startup requirements for nuclear power (10 seconds and

NOTES:

THIS ANALYSIS IS BASED ON

1- 800 MW LIGHT WATER REACTOR
1- 500 MW FOSSIL FIRED STEAM
5- 20 MW HIGH SPEED GAS TURBINES
3- 33 MW HEAVY DUTY GAS TURBINES
20- 5 MW DIESEL PEAKING UNITS
12- 6.4 MW MEDIUM SPEED DIESELS
8-12.5 MW LOW SPEED DIESELS

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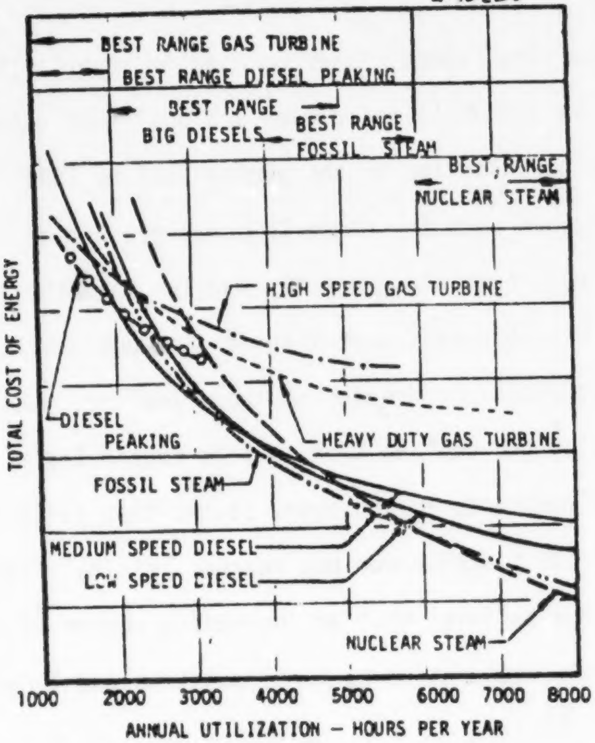

Figure 8-5. Chart shows relative cost data and most favorable operating ranges for the various types of generating facilities, as derived from one particular study (from Reference 23 -- costs for 1968).

load capability in 30 seconds), this service is almost exclusively met by large diesel engines. Since safety regulations require that there be at least two engines for each reactor, the best indicator of future engine needs by this market is a record of scheduled construction of nuclear power reactors. Table 8-6 shows the number of reactors scheduled for completion[1/]. This indicates a market for 338 to 382 high-power engines in the next 10 to 15 years. Although recent difficulties in raising capital and in proving the safety of reactors and spent fuel disposal have caused utilities to delay over 40 percent of the units under construction or on order and to cancel 5 to 10 percent, the Nuclear Regulatory Commission (formerly the Atomic Energy Commission) continues to project that 102,000 nuclear megawatts will be constructed by 1980 and 250,000 by 1985(30). It should be recognized that the engines for a particular reactor may be purchased up to a year or two before the reactor becomes operational.

TABLE 8-6. PLANNED CONSTRUCTION OF NUCLEAR REACTORS

|  | Completed | Scheduled as of January 1, 1974 |  |  |  |  |  |
|---|---|---|---|---|---|---|---|
|  | During 1973 | 1974 | 1975 | 1976 | 1977 | 1978 | 1979 & later |
| Reactors | 7 | 27 | 10 | 7 | 12 | 11 | 102 |

Radar power stations are also served by reciprocating engines which maintain precise power characteristics over sizable load variations.

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[1/] The source of this table(28) lists 169 units completed or scheduled for completion after 1973. Another source mentions that 191 units are currently under construction or on order.

Engines have also been in demand by flood control districts for pumping applications along the Mississippi Delta.

### 8.1.3.2 Markets for Dual-Fuel Engines

The concept of dual fuel operation was developed to take advantage of both compression ignition performance and inexpensive natural gas. These engines have been used almost exclusively for prime electric generation. Figure 8-6(31-34) illustrates, however, that shortages of natural gas and the 1973 oil embargo have combined to significantly reduce the sales of these engines in recent years. For example, sales of dual-fuel engines in 1971, 1972, 1973, 1974, and 1975 were 95, 74, 53, 17, and 35 units, respectively(35,36). As discussed above, spokesmen for engine manufacturers stated that sales have recovered somewhat as demands for power have become critical and firm commitments for fuel are established.

### 8.1.3.3 Markets for Natural Gas Engines

The primary application of large gas engines during the past 5 years has been for oil and gas production. Figure 8-7(37-42), based on manufacturer's data from responses to the June 16, 1976 Section 114 Request for Information, illustrates that 75 to 80 percent of all gas engine horsepower sold during the past 5 years was used for this application. The primary uses are to power gas compressors for recovery, gathering, and distribution.

During this time, sales to pipeline transmission applications declined. Combined with standby power, electric generation, and other services (industrial and sewage pumping), these applications accounted for the remaining 20 to 25 percent of horsepower sales. The growth of oil and gas production applications during this period corresponds to the

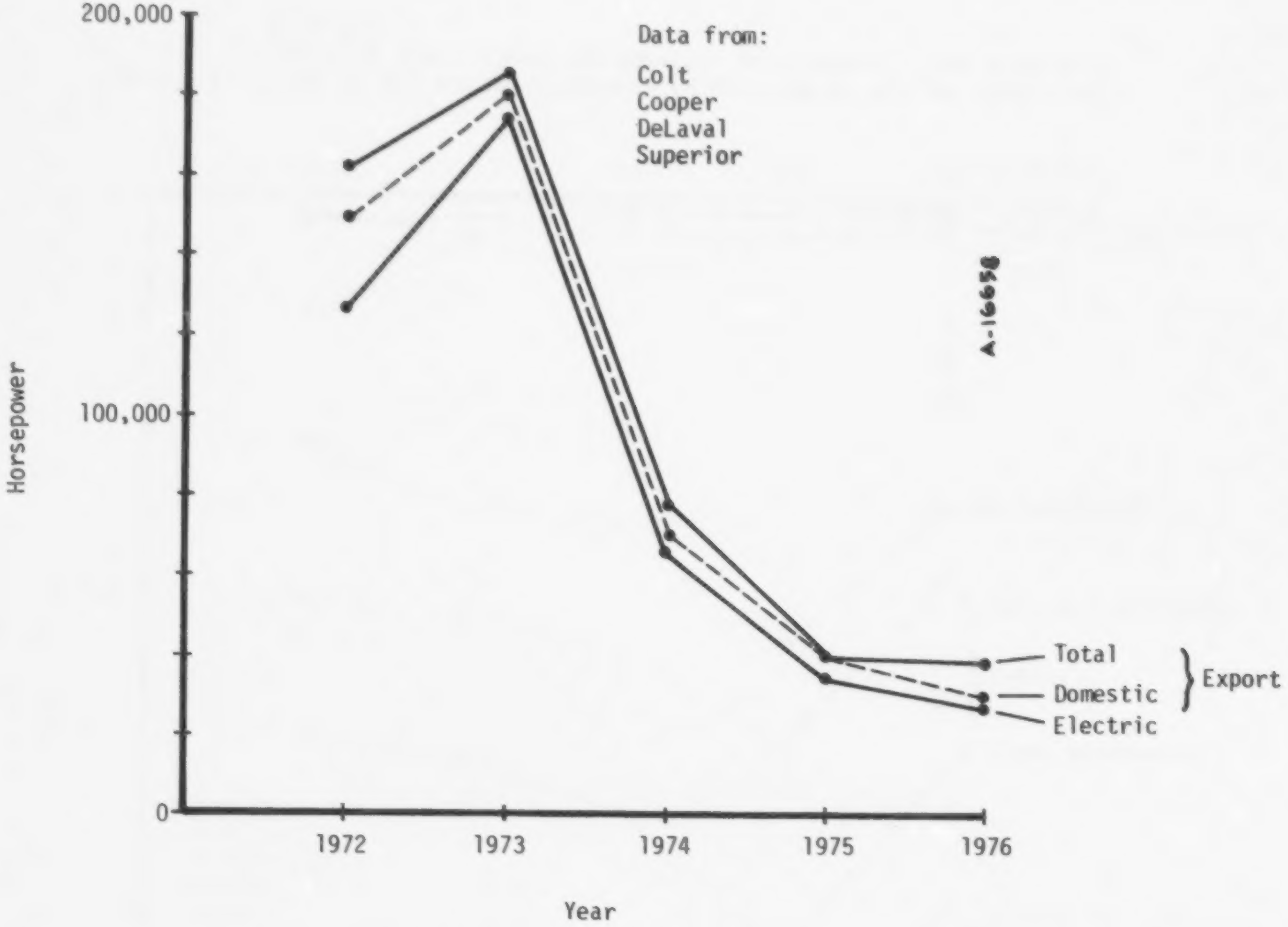

Figure 8-6. Sales of dual-fuel horsepower during the past 5 years (from References 31 to 34).

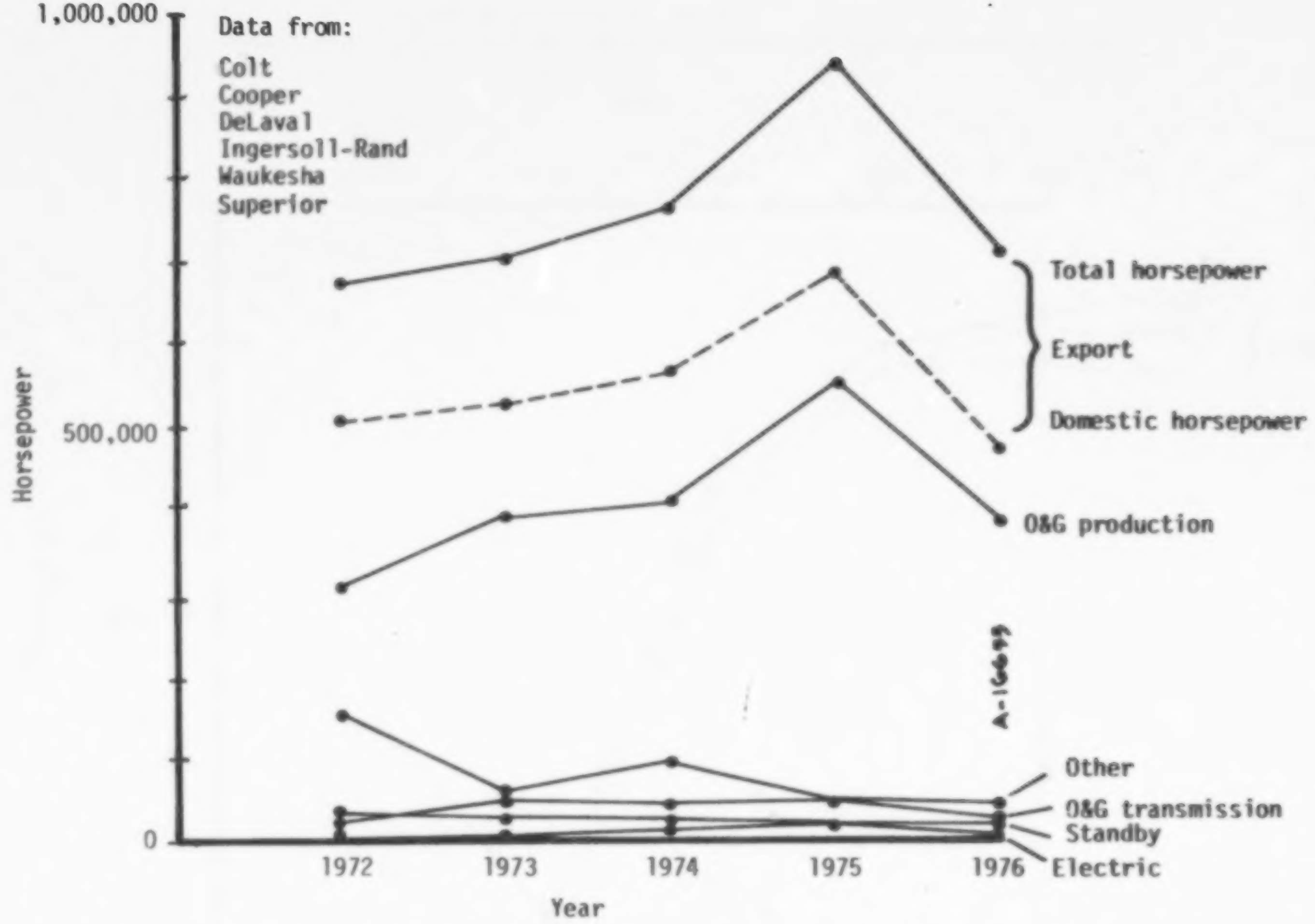

Figure 8-7. Sales of gas engine horsepower by application for the past 5 years (absolute levels shown for domestic applications, from References 37 to 42).

increasing efforts to find new, or recover marginal, gas reserves, and distribute them to our existing pipeline transmission network, and store in covered, underground reservoirs near cities for peak winter demands.

Figure 8-8 illustrates the number of gas engines sold for five size groups during the past 5 years. The large number of smaller than 500-hp engines that were sold during this period are primarily one or twocylinder engines used on oil well beam pumps and for natural gas well recovery and gathering. Most of the other, larger gas engines that were sold during this period ranged from 500- to 2000-hp. In 1976, approximately 400 engines in this size range were sold, primarily for oil and gas production (see Figure 8-7). Most of these gas engines were manufactured by Caterpillar, Cooper, Waukesha, and Superior Division of Cooper.

Historical sales data for pipeline transmission and field compressor stations (see Figure 8-9)(43) clearly indicate the recent market position for IC engines and gas turbines. Total sales of IC engines have been relatively constant since 1970, while total gas turbine sales have decreased dramatically with the recent slowdown of new pipeline construction. A breakdown of sales for transmission and field applications in 1975 (year ending June 15) is given below(44):

| Prime Mover | Turbine | | Engine | |
|---|---|---|---|---|
| Compressor Horsepower | Transmission Stations | Field Stations | Transmission Stations | Field Stations |
| New | 21,933 | 2,000 | 4,080 | 29,400 |
| Additions | 23,300 | 3,500 | 78,800 | 30,450 |
| Total | 45,233 | 5,500 | 82,880 | 59.850 |

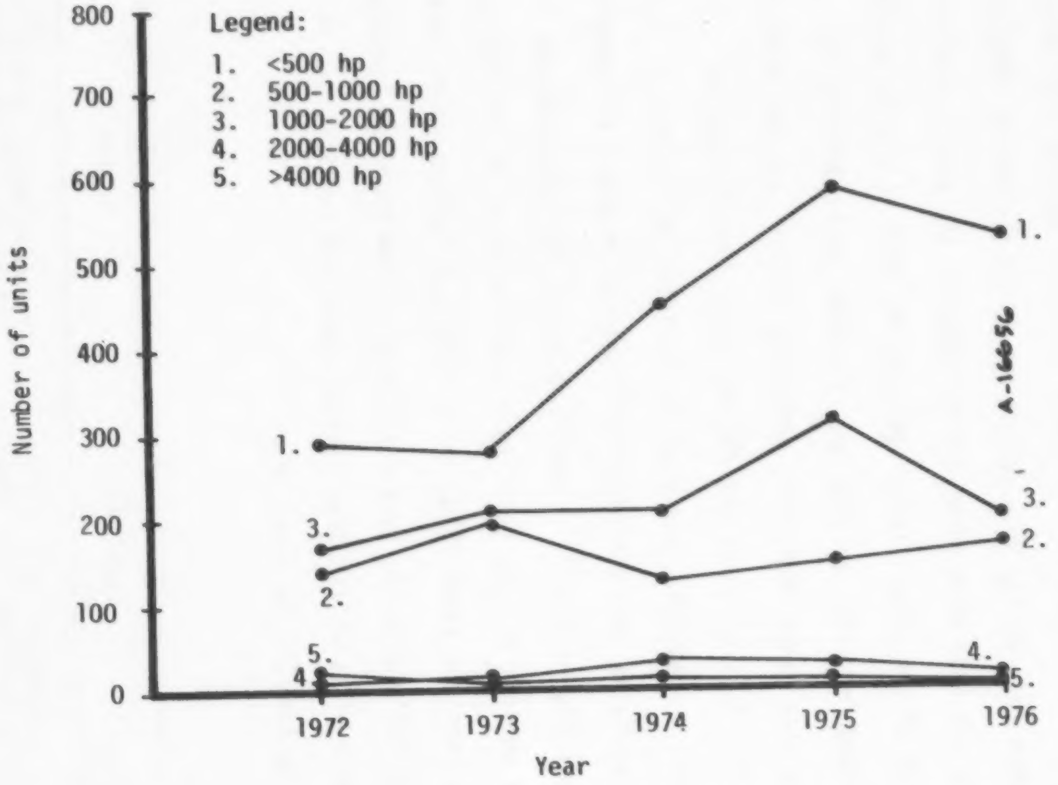

Figure 8-8. Size distribution of gas engines sold during the past 5 years.

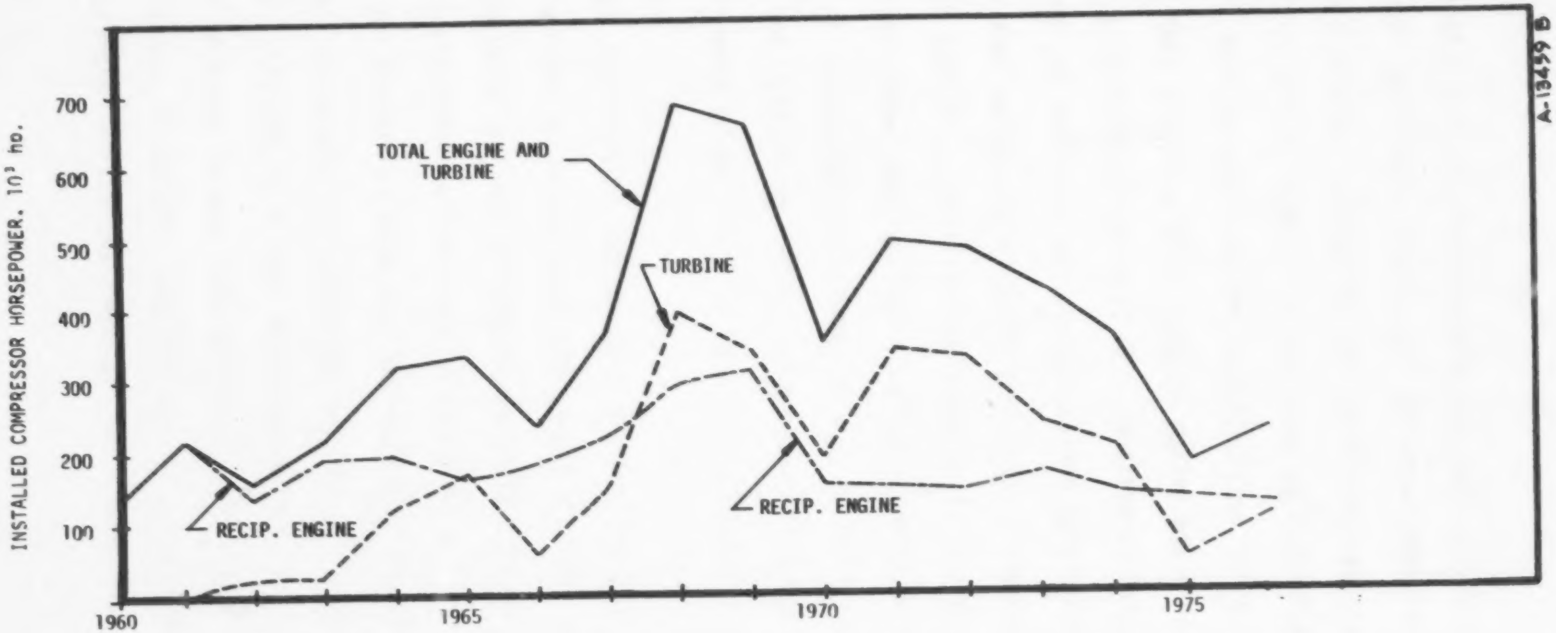

Figure 8-9. New and added compressor capacity on gas transmission pipelines (Reference 43).

These data indicate that new transmission pipeline projects are almost exclusively equipped with turbine-powered centrifugal compressors, while new field stations (gathering, recompression, storage) are powered by engine-driven reciprocating compressors. Widely varying loads are encountered in these latter applications, and therefore, engine-driven reciprocating compressors are better adapted to this service than are turbine-driven rotating compressors. In 1975 nearly 80 percent of the total sales of reciprocating horsepower was for additions to, or replacement of, existing compressor stations, while turbine sales were evenly divided between new and existing compressor stations. In some cases the gas turbine sales to existing stations have displaced reciprocating engines. This has occurred in growing compressor stations or in facilities where all the old engines reach retirement age at the same time and can be replaced more cheaply by one large turbine than by several reciprocating engines.

Engines used in pipelines are concentrated in the major gasproducing areas, such as the Gulf Coast, and along the major natural gas pipelines. Pipeline construction has dropped in the last several years, but applications to the FPC for pipeline construction increased during 1973 and the first half of 1974 as plans were made to exploit the natural gas discovered in Alaska(45). If the percentage of compressor stations utilizing reciprocating engines remains the same as in the past, this will bring an increase in engine sales over the next several years as pipeline companies purchase compressors to move the gas. The exact impact is uncertain, however, as firm orders for equipment await final approval by the FPC.

According to one major western oil company, in the past 10 years there has been a movement away from reciprocating engines used in refinery operations to electric motors, steam turbines for high-load requirements, and, occasionally, fossil-fueled turbines(46). Moreover, a plant manager from one of the major chemical processing firms, who utilizes large reciprocating engines to compress gases, believes that large power users will purchase gas turbines in the future for fuel conservation reasons. Most of these users can utilize the waste heat from the prime mover, and more of this energy can be recovered economically from one very large turbine than from several large engines(47).

Municipalities use high-power, spark-ignited engines to generate electricity from digester gas in sewage treatment plants and to pump water. Competition for these applications comes from gas turbines and electric motors. As with electric power generation, IC engines have an advantage over turbines for applications when fuel costs are a significant portion of annual costs. Engines are preferred over electric motors in areas where electricity is relatively more expensive than liquid or gaseous fuels and in applications such as sewage processing where a by-product (such as sewage gas) can be burned to supplement other fuels.

### 8.1.4 Balance-of-Trade

The U.S. Bureau of the Census (Department of Commerce) does not classify imported and exported IC engines into stationary and nonstationary applications. Furthermore, there are no priority reasons for assigning a breakdown by application to either imported or exported engines. Therefore, information on the balance-of-trade for the stationary engine market is limited to the following categories, for which the Department of Commerce does report data:

| | |
|---|---|
| Imports: | Engines for agricultural machines; compression ignition engines; aircraft engines; bus, auto, truck engines; outboard motors; and engines not elsewhere classified (NEC) |
| Exports: | Diesel engines (automotive, marine, and NEC); gas engines other than turbines; outboard engines; gasoline engines (marine, automotive, and NEC); and IC engines NEC |

Imported stationary engines would be included in the "Compression Ignition Engines," "Agricultural Machines," and "Not Elsewhere Classified" categories. These categories also include engines used for marine and construction applications. For this reason it is not possible to determine the exact number of imported stationary engines or the exact impact stationary source regulations would have on imports.

The classifications are less of a problem for exports because the categories are more narrowly defined. Furthermore, emission regulations on future domestic engines would only affect exported engines if the U.S. manufacturers added control devices to all their engines produced in the U.S. rather than maintaining two lines of engines -- one for regulated engines and one for unregulated engines. Table 8-7(48,49) gives import and export data for the appropriate categories of IC engines during the fiscal years 1969 to 1973.

Except for 1972, the trade balance for IC engines has been positive at about $60 million to $75 million per year and is improving. It is interesting to note that the average value of exported diesel engines is about six-times the average value of imported diesel engines, while the average value of exported gasoline engines is about one-third the average value of imported gasoline engines. Using 1971 Commerce Department price

## TABLE 8-7. IMPORT AND EXPORT OF IC ENGINE CATEGORIES THAT INCLUDE STATIONARY APPLICATIONS[a]

| | | Exports | | | Imports[b] | | | Balance-of-Trade[c] ($1000) |
|---|---|---|---|---|---|---|---|---|
| | | Quantity | Value ($1000) | Average Value ($) | Quantity | Value ($1000) | Average Value ($) | |
| Gasoline Engines | 1969 | 1,000,911 | 52,858 | 52 | 276,438 | 25,795 | 93 | 28,429 |
| | 1970 | 858,688 | 49,964 | 56 | 325,696 | 36,432 | 112 | 14,700 |
| | 1971 | 761,912 | 42,375 | 56 | 425,779 | 41,153 | 97 | 6,097 |
| | 1972 | 976,601 | 51,930 | 53 | 620,925 | 57,524 | 92 | (1,351) |
| | 1973 | 1,703,460 | 83,673 | 49 | 330,224 | 49,190 | 148 | 37,168 |
| Diesel Engines | 1969 | 19,221 | 64,677 | 3,365 | 48,413 | 30,609 | 632 | 34,068 |
| | 1970 | 19,439 | 85,527 | 4,400 | 48,511 | 36,834 | 759 | 48,693 |
| | 1971 | 16,555 | 101,135 | 6,109 | 43,280 | 33,794 | 780 | 67,341 |
| | 1972 | 17,156 | 83,397 | 4,861 | 55,765 | 47,892 | 860 | 35,505 |
| | 1973 | 20,968 | 104,760 | 4,996 | 72,271 | 64,289 | 889 | 40,471 |
| Gas Engines | 1969 | 769 | 1,366 | 1,776 | -- | -- | -- | -- |
| | 1970 | 1,009 | 1,168 | 1,157 | -- | -- | -- | -- |
| | 1971 | 1,436 | 4,865 | 3,387 | -- | -- | -- | -- |
| | 1972 | 833 | 4,243 | 5,093 | -- | -- | -- | -- |
| | 1973 | 988 | 2,685 | 2,717 | -- | -- | -- | -- |
| Total | 1969 | 1,020,901 | 118,901 | | 324,851 | 56,404 | | 62,497 |
| | 1970 | 879,136 | 136,659 | | 374,207 | 73,256 | | 63,393 |
| | 1971 | 779,903 | 148,375 | | 469,059 | 74,947 | | 73,428 |
| | 1972 | 994,590 | 139,570 | | 676,690 | 105,416 | | 34,154 |
| | 1973 | 1,725,416 | 191,118 | | 402,495 | 113,479 | | 77,639 |

[a] These categories are Compression Ignition Engines and Engines NEC for imports; they are Diesel Engines NEC, Gas Engines other than Turbines, and Gasoline Engines NEC for exports (References 48, 49). These categories also include an unknown number of engines for mobile use.

[b] Gas engine imports are included in the figures for gasoline engine imports.

[c] Balance-of-Trade for gasoline and gas engines. Number in parenthesis means a deficit.

data for the value of engines produced in the U.S. as a function of their rated power, the following information can be derived for 1973 imports and exports(50).

| | Diesel Engines | | Gasoline Engines | |
|---|---|---|---|---|
| | Imports | Exports | Imports | Exports |
| Average value per engine, $ | 889 | 4996 | 148 | 49 |
| Corresponding average hp | 60 | 300 | 11 | 6 |

Thus, future imports of diesel engines should correlate with U.S. demand for small diesel engines. Most of the diesel-powered portable refrigeration units and underground mining machinery in the U.S. use imported diesel engines (51,52).

Since international markets for all capital equipment are highly competitive, trade balances of engines may be affected more by monetary exchange rates and tariff restrictions than by price changes due to emission control systems. Moreover, based on the average horsepower shown above, large-bore engines play an insignificant role in the import or export market of stationary reciprocating IC engines.

## 8.2 COST ANALYSIS FOR CONTROL OF $NO_x$ EMISSSIONS

This section presents a discussion of the cost impact to the engine user and manufacturer of implementing the viable $NO_x$ control options designated in Chapter 6. The costs to the engine user of purchasing and operating engines equipped with selected $NO_x$ controls are discussed in Section 8.2.1. The costing was done on the basis of information supplied by the manufacturers and users, and is applied to a group of "model" engines that are typical of those used in a particular application.

In Section 8.2.2 the costs to the manufacturers for the implementation of the alternative $NO_x$ controls (presented in Chapter 6) are discussed. These cost considerations include additional manufacturing associated with adaptation of controls to existing designs and the costs of engineering, tooling, and verifying the effectiveness of a particular control approach.

Section 8.2.3 presents those costs associated with emerging control techniques. In Sections 8.2.4, 8.2.5 and 8.2.6 costs associated with fuel pretreatment, modified facilities, and reconstructed facilities, respectively, are identified.

### 8.2.1 New Engines

The application of $NO_x$ controls will affect costs to the engine manufacturer and the engine user. The degree of the effect will depend upon both the amount of reduction applied and the type of control applied. As was shown in Section 6.3, various control approaches affect initial costs, fuel consumption, and maintenance differently. Furthermore, manufacturers of stationary engines may incur different costs to achieve a given $NO_x$ reduction depending on a number of factors including: (1) their degree of advancement in emissions testing, (2) the uncontrolled emission rates of their engines, and (3) the necessary R&D required to produce engines which can meet proposed standards of performance. Therefore, the discussion of incremental costs to employ $NO_x$ controls for manufacturers and engine users will be treated separately. This section will be restricted to the discussion of incremental costs incurred by engine users, and Section 8.2.2 will discuss $NO_x$ costs related to engine manufacturers.

To illustrate the effect of $NO_x$ controls on costs, "model" engines will be selected to represent major end users of diesel, dual-fuel, and natural gas engines. Baseline costs, comprised of investment and operating expenses, will be established for each model. Computations for these model units will then be used to illustrate the range of incremental costs to the user resulting from the application of the $NO_x$ control systems described in Section 6.2. These incremental costs will be illustrated for several control systems that achieve any one of three levels of $NO_x$ reduction (20 percent, 40 percent, and 60 percent). This approach is not intended to be a comprehensive cost analysis of all possible $NO_x$ control systems; rather it is intended to illustrate a range of costs that an engine user would incur to achieve a given level of $NO_x$ reduction. The discussions are subdivided by major end uses, since engine types and costs are unique to each end use.

Section 8.2.1.1 briefly describes the models selected to represent major engine applications. The cost analysis methodology is then discussed in Section 8.2.1.2, and the results of the cost analysis are presented in Section 8.2.1.3.

### 8.2.1.1 Model Engines

Four model engines have been selected to represent the major applications of diesel, dual-fuel, and natural gas engines. (The applications were described in Section 8.1.3) The following paragraphs briefly describe these models.

**Diesel Engine Model: Electrical Generation**

As described in Section 9.3.1, affected diesel engines are large-bore, exceeding a displacement of 560 cubic inches per cylinder. Typically these engines are used as prime movers for electrical generators in municipal utilities. These engines operate from 6000 to 8000 hrs/year (baseload) and consume approximately 7000 Btu/hp-hr of operation. Manufacturers of these engines include Alco, Colt, Cooper and Superior Division (of Cooper), DeLaval, and ElectroMotive division of General Motors.

**Dual-Fuel Engine Model: Electrical Generation**

These engines are nearly identical to the diesel engines except that they burn predominantly natural gas (typically 95 percent of the total fuel heating value). In general, these engines operate more efficiently than their diesel counterparts, consuming 6500 Bth/hp-hr of operation. Manufacturers of affected dual-fuel engines include Colt, Cooper and Superior Division (of Cooper), and DeLaval.

**Gas Engine Model: Oil and Gas Transportaion**

These engines are installed on pipeline compressors for long-range transportation of natural gas. They generally exceed 1000 hp, averaging 3000 to 4000 hp. Typical annual usage is 8000 hrs, and representative fuel consumption is 7000 Btu/hp-hr. Manufacturers of gas engines for this application include Colt, Cooper and Superior Division (of Cooper), DeLaval, and Ingersoll-Rand.

Gas Engine Model: Oil and Gas Production

These engines are installed on compressors that gather, store, process, or distribute gas from gas production fields. (These engines generally burn gas that has been treated to reduce the sulfur content.) As described in Section 9.3.3, engines that would be affected by proposed standards of performance range from 300 to 2000 hp. An average size engine is about 1000 hp and consumes about 8000 Btu/hp-hr. These engines are manufactured primarily by Caterpillar, Cooper and Superior, Waukesha, and Ingersoll-Rand.

### 8.2.1.2 Costing Model for Users of Stationary IC Engines

The objective of the cost analysis is to estimate how an engine user's life cycle costs will change with the application of $NO_x$ controls for the model diesel, dual-fuel, and gas engines described in Section 8.2.1.1. The following paragraphs describe the cost analysis approach and the basic assumptions that are used to estimate the incremental costs created by $NO_x$ controls. Section 8.2.1.3 will then present the results of the cost analysis.

Methodology

The costs of owning and operating a large-bore engine can be represented as follows:

$$TAC = AIC + M + F$$

where TAC = total annual cost of ownership and operation of engine

AIC = annualized initial cost = initial engine cost x capital recover factor (CRF)

M = maintenance costs

F = fuel and lubrication costs

The annualized initial cost includes capital recovery of the initial investment (assuming 100 percent debt financing), depreciation, property taxes, and insurance. Capital recovery rates typically range from 15 to 25 percent. Conversations with industry spokesmen indicate that a rate of 20 percent is appropriate[2/] for estimating installed engine costs for electrical generation, gas production, and gas transportation applications(53,54).

The procedure for computing the incremental costs of various $NO_x$ control techniques is as follows:

- Estimate the increase in the costs of AIC, M, and F due to $NO_x$ controls
- Compute the increase in the total annual cost, TAC
- Present the results as $(TAC_c - TAC_u)/TAC_u \times 100$ = percent increase in TAC where c = controlled and u = uncontrolled

## Basic Costs and Parameters for Cost Analysis

Table 8-9 summarizes the basic inputs for computing the total annual costs of uncontrolled engines. As this table indicates, the costs will be presented in a brake specific format, that is, in \$/hp-hr. The initial costs are normalized by the output power and usage rate to obtain \$/hp-hr. This format makes it possible to compare ownership costs for a number of differently sized engines that are used in the same applications. This format also permits a direct comparision of the incremental $NO_x$ control costs among engines using different fuels.

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[2/] CRF computed assuming a 30-year physical life, 20-year accounting life, 100-percent debt financing, 10-percent interest on debt, and 4-percent fixed capital expense (e.g., property tax, insurance, and direct overhead). On these assumptions, debt service is 10.6 percent, depreciation 5 percent, and fixed capital expense 4 percent a total of 19.6 percent.

As shown in Table 8-8, typical initial costs for diesel and dual-fuel electrical generation and oil and gas transmission engines are \$150[(55)]. (This cost is for the engine only, F.O.B..) Costs for gas production engines are estimated at \$50/hp and are representative of engines sold by Caterpillar and Waukesha[(56)]. Total capital investments for installed electrical generation stations are approximately \$300/hp (1976)[(57)]. Current investments for installed gas transmission compressor installations range from \$318 to \$575/hp, and range up to \$584/hp for gas field stations[(58)]. Thus, the cost increases computed in this section will be considerably smaller when expressed as a percentage of the total application investment.

Maintenance costs for these engines have been estimated based on information supplied by engine manufacturers. These costs are typical of engines that operate continuously at rated load.

Fuel costs assumed for the electrical generation applications are representative of costs of distillate oil and natural gas transported both intra- and interstate. (The current (1978) regulated price of interstate gas is \$1.48 per Mcf.) Lower gas costs have been assumed for oil and gas production and transportation applications since gas companies own these facilities and pay less for the gas. In addition, average gas costs for these companines are a composite of contracted supplies of gas that span several years[(59)]. Fuel consumption estimates are average values based on the data presented in Section 4.3.1. Note that this analysis assumes baseload or continuous annual operation (≈8000 hr/yr). Figure 8-9 illustrates the relative proportion of each of these items relative to total uncontrolled costs. Fuel and lubrication costs are the largest fraction

## TABLE 8-8. BASELINE COST PARAMETERS FOR LARGE-BORE ENGINES

| COST PARAMETERS | ENGINE MODELS | | | |
|---|---|---|---|---|
| | DIESEL (Electrical Generation) | DUAL-FUEL (Electrical Generation) | NATURAL GAS | |
| | | | (Oil & Gas Transport) | (Oil & Gas Production) |
| Initial Cost[a,b], \$/hp | 150 | 150 | 150 | 50 |
| Capital Recovery Factor | 0.2 | 0.2 | 0.2 | 0.2 |
| Annual Usage, hr/yr | 8000 | 8000 | 8000 | 8000 |
| Maintenance[a] | | | | |
| Parts, \$/hp-hr | 0.0012 | 0.0012 | 0.0012 | 0.0012 |
| Labor, \$/hp-hr | 0.0018 | 0.0018 | 0.0018 | 0.0018 |
| Total, \$/hp-hr | 0.0030 | 0.0030 | 0.0030 | 0.0030 |
| Fuel Cost[c], $\$/10^6$ Btu | 2.50 | 3.00 | 2.00 | 2.00 |
| Fuel Consumption, Btu/hp-hr | 7000 | 6500 | 7000 | 8000 |
| Lubrication[a], % Fuel Cost | 5 | 10 | 10 | 10 |

[a] Aggregated from confidential communications with engine manufacturers.

[b] Initial cost divided by annual usage (8000 hr) and multiplied by the capital recovery factor (0.2) gives annualized cost in \$/hp-hr.

[c] Fuel cost calculations for dual-fuel units assume fuel consumption is 100% gas. All fuel costs based on 1977 information.

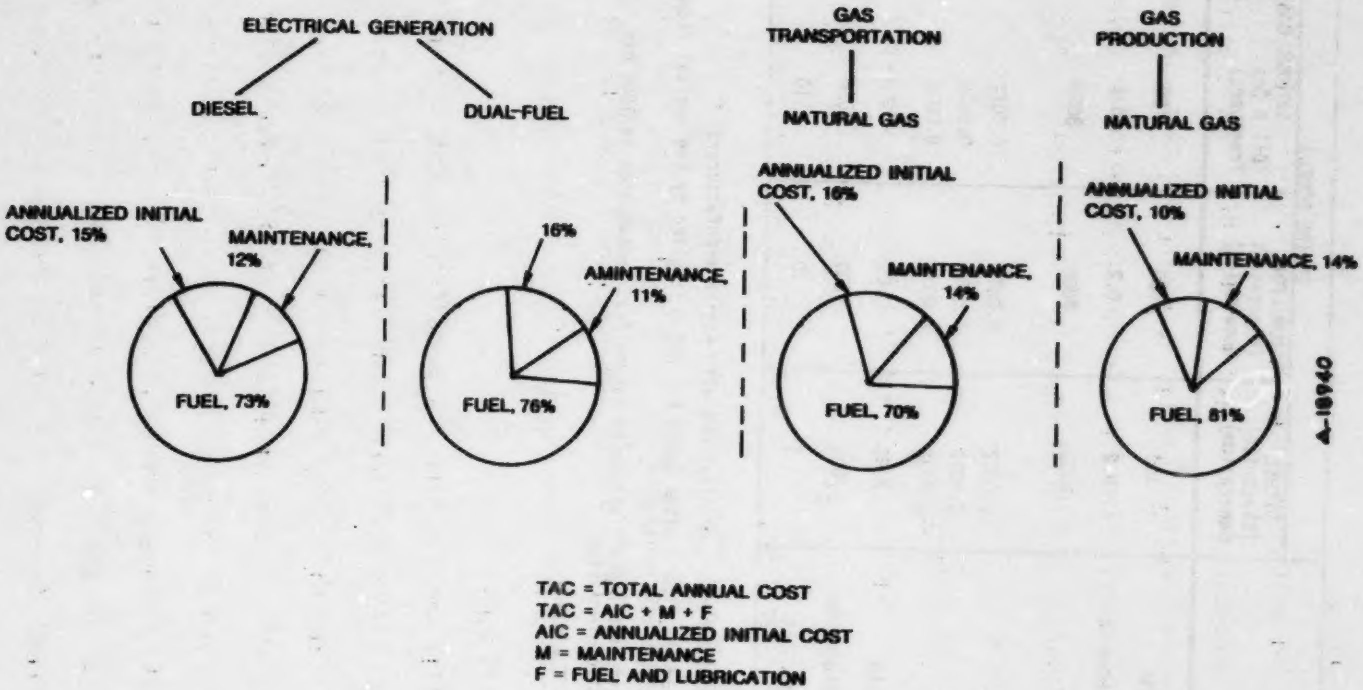

Figure 8-10. Relation of initial maintenance and fuel costs to total costs for major applications of large-bore engines.

of total costs, accounting for 70 to 80 percent of the total. The variation is a result of the different engine and fuel costs assumed for each of the model applications.

The capital and maintenance cost increases resulting from the application of $NO_x$ controls are estimated in Table 8-9. These estimates were aggregated from information supplied by engine manufacturers or vendors of auxiliary equipment. Increases in fuel consumption will be estimated for various control techniques based on the information given in Table 6-4 of Section 6.2.

### 8.2.1.3 Results of Cost Analysis

A cost analysis based on the preceding discussion was performed for each of the four model engines described in Section 8.2.1.1. Total annual cost penalties (as a percentage of total uncontrolled costs) were computed for each model engine and alternative level of $NO_x$ reduction (i.e., 20-, 40-, and 60-percent reduction) for the control techniques that were discussed in Section 6.2. Table 6-4 is repeated here as Table 8-10 to illustrate those techniques and their fuel penalties for each level of control alternative and fuel type.

Fuel penalties are the major factor affecting cost increases for high usage engines. Table 8-10 shows that fuel penalties increase with increasing level of control. They also vary with control type. For example, derating results in substantial penalties (>10 percent) for $NO_x$ reductions greater than 20 percent. Retard, manifold air cooling, and air-to-fuel controls, however, generally achieve $NO_x$ reductions at a penalty less than 10 percent. It should be noted that the decrease in data at the 60 percent $NO_x$ reduction level is the result of both: (1) manufacturer's inexperience with the application of controls to the extent

## TABLE 8-9. COST INCREASES TO THE ENGINE USER RESULTING FROM $NO_x$ CONTROL[a]

| CONTROL | CAPITAL | MAINTENANCE |
|---|---|---|
| Retard | None | 33% increase in base cost due to 25% reduced service life of exhaust valves for dual-fuel engines |
| Air-to-Fuel | None | Increase of 0.0001/hp-hr for increased cleaning of turbochargers |
| Derate | Increase by ratio of rated power to derated power (to compensate for power loss) | Increase by ratio of rated power to derated power (more units or cylinders to service) |
| Manifold Air Temperature Reduction | Increase engine costs 1.5% to acheive 100°F inlet air. Larger heat exchangers cost assumes engine equipped with intercooler | D, DF: Increase $0.0005/hp/hr for cooling water chemical treatment (cooling towers)[b]<br>G: Increase $0.0001/hp-hr for increased service of radiator and aftercooler[c] |
| External Exhaust Gas Recirculation | Increase engine cost 5% for plumbing, heat exchanger and controls for 10-12% recirculation. | Double parts for diesel engines[d]<br>Triple parts for dual-fuel and gas engines[d] |

[a] Aggregated from confidential communications with engine manufacturers.

[b] Cooling water from cooling towers for diesel and dual-fuel engine must be treated to prevent sludge and scale buildup due to water "hardness".

[c] Closer tolerances to achieve lower manifold air temperature will require more frequent cleaning and servicing of radiators and intercoolers (or aftercoolers) of gas engines.

[d] Diesel unit has fixed rate of EGR, dual-fuel, and gas units have a variable rate of EGR. Charge for parts includes periodic replacement of the EGR system and its controls.

## TABLE 8-10. $NO_x$ CONTROL TECHNIQUES THAT ACHIEVE SPECIFIC LEVELS OF $NO_x$ REDUCTION

T-654

| $NO_x$ Reduction | Diesel | | Dual Fuel | | Natural Gas | |
|---|---|---|---|---|---|---|
| | Control (amount) | ΔBSFC,[a] % | Control (amount) | ΔBSFC,[a] % | Control (amount) | ΔBSFC,[a] % |
| 20% | Retard (2 to 4°) | 0 to 4 | Retard (2 to 3°) | 1 to 3 | Retard (4 to 5°) | 1 to 4 |
| | External EGR (7%) | 0 | Manifold air cooling | 1 | Manifold air cooling | 0 |
| | Derate (25 to 50%) | 3 to 5 | External EGR (10%) | 1 | External EGR (4%) | 0 |
| | Air-to-fuel change (25%) | 10 | Derate (12 to 25%) | 0 to 8 | Derate (5 to 35%) | 2 to 6 |
| | Retard & manifold air cooling | 0 to 1 | Air-to-fuel changes (5 to 10%) | 0 to 2 | Air-to-fuel change (5%) | 0-2 |
| | Retard & manifold air cooling & air-to-fuel change | | | | | |
| 40% | Retard (7 to 8°) | 4 to 8 | Retard (5°) | 2 | Retard (10°) | 2 |
| | Derate (50%) | 14 to 17 | Manifold air cooling | 1 | Derate (10 to 50%) | 2 to 24 |
| | Air-to-fuel change & manifold air cooling | 3 to 5 | Derate (30 to 50%) | 4 to 8 | Air-to-fuel change (7%) | 2 |
| | Retard & air-to-fuel change | | Air-to-fuel change (10%) | 2 | Retard & manifold air cooling & air-to-fuel change | 7 |
| | | | Retard & manifold cooling | 3 | | |
| 60% | Retard (16°) | 19 to 24 | Retard (6°) | 2 | Derate (10 to 50%) | 2 to 22 |
| | Retard & air-to-fuel change | 21 | Derate (50%) | 12 | Air-to-fuel change (8 to 12%) | 2 to 5 |
| | | | Retard & air-to-fuel change | 1 to 3 | Retard & manifold air cooling & air-to-fuel change | 7 |

[a] ΔBSFC = increase in brake specific fuel consumption.

necessary to achieve that level, and (2) in some cases the inability of a particular control approach to achieve reductions at this level.

The differential control costs for the techniques shown on Table 8-10 are tabulated in Tables 8-11 to 8-14 for the four end use applications described above. Table 8-15 is a summary of these cost penalties. In general, retard, manifold air cooling, air-to-fuel change, or some combination of these achieved 60-percent $NO_x$ reductions for less than a 10-percent cost penalty for each application except diesel/electric generation. The cost penalty for the diesel/electric generation, 60-percent $NO_x$ reduction category, however, is based on data from tests of only one engine model; therefore this result may not be representative of costs for other engine models.

The data in Table 8-15 indicate a wide variation in cost penalty at any level of $NO_x$ reduction. Moreover, average cost penalties are less than 6 percent (with the exception of diesel engines) for a 60-percent $NO_x$ reduction. Nevertheless, average cost penalties increase as the level of $NO_x$ reduction increases.

Since average uncontrolled $NO_x$ emission rates from engines of different manufacturers vary, cost penalties to achieve a given alternative performance standard will also vary among manufacturers. These differential costs are important to identify so that potential economic impacts in various end use markets can be identified (see Section 8.4.1).

Table 8-16 illustrates the cost penalties for each manufacturer and fuel type for each of the three alternative levels of performance standards. In general, the maximum cost penalty for any fuel is less than 10 percent with the exception of the 40- and 60-percent reduction levels for diesel engines. The data for gas engines do not indicate differential

## TABLE 8-11. COST OF ALTERNATIVE NO$_x$ CONTROLS FOR DIESEL ENGINES FOR ELECTRICAL GENERATION

| NO$_x$ Control % Reduction | | Cost Component[a] | Retard (R) | | Derate (D) | | External EGR | | Air-to-Fuel (A) | | R+M | | R+M+A | | R+A | | A+M | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | $/hp-hr | $/hp-hr | %Chg | $/hp-hr | %Chg | $/hp-hr | %Chg | $/hp-hr | %Chg | $/hp-hr | %Chg | $/hp-hr | %Chg | $/hp-hr | %Chg | $/hp-hr | %Chg |
| 20% | Low | AIC | 0.00375 | 0 | 0.005 | 33 | 0.00394 | 5 | 0.00375 | 0 | 0.00381 | 1.5 | 0.00381 | 1.5 | | | | |
| | | M | 0.003 | 0 | 0.004 | 33 | 0.0042 | 40 | 0.0031 | 3 | 0.0035 | 17 | 0.0035 | 17 | | | | |
| | | F | 0.01838 | 0 | 0.01893 | 3 | 0.01838 | 0 | 0.02022 | 10 | 0.01838 | 0 | 0.01856 | 1 | | | | |
| | | TAC | 0.02513 | 0 | 0.02738 | 9 | 0.02652 | 6 | 0.02707 | 8 | 0.02569 | 2 | 0.02587 | 3 | | | | |
| | High | AIC | 0.00375 | 0 | 0.0075 | 100 | | | | | | | | | | | | |
| | | M | 0.003 | 0 | 0.0060 | 100 | | | | | | | | | | | | |
| | | F | 0.01912 | 4 | 0.01930 | 5 | | | | | | | | | | | | |
| | | TAC | 0.02587 | 3 | 0.03280 | 31 | | | | | | | | | | | | |
| 40% | Low | AIC | 0.00375 | 0 | 0.0075 | 100 | | | | | | | | | 0.00375 | 0 | 0.00381 | 1.5 |
| | | M | 0.003 | 0 | 0.006 | 100 | | | | | | | | | 0.0031 | 3 | 0.0035 | 17 |
| | | F | 0.01912 | 4 | 0.02095 | 14 | | | | | | | | | 0.01930 | 5 | 0.01893 | 3 |
| | | TAC | 0.02587 | 3 | 0.03445 | 37 | | | | | | | | | 0.02615 | 4 | 0.02624 | 4 |
| | High | AIC | 0.00375 | 0 | 0.0075 | 100 | | | | | | | | | | | | |
| | | M | 0.003 | 0 | 0.006 | 100 | | | | | | | | | | | | |
| | | F | 0.01985 | 8 | 0.02150 | 17 | | | | | | | | | | | | |
| | | TAC | 0.02660 | 6 | 0.03500 | 40 | | | | | | | | | | | | |
| 60% | Low | AIC | 0.00375 | 0 | | | | | | | | | | | | | | |
| | | M | 0.003 | 0 | | | | | | | | | | | | | | |
| | | F | 0.02187 | 19 | | | | | | | | | | | | | | |
| | | TAC | 0.02862 | 14 | | | | | | | | | | | | | | |
| | High | AIC | 0.00375 | 0 | | | | | | | | | | | 0.00375 | 0 | | |
| | | M | 0.003 | 0 | | | | | | | | | | | 0.0031 | 3 | | |
| | | F | 0.02279 | 24 | | | | | | | | | | | 0.02224 | 21 | | |
| | | TAC | 0.02954 | 18 | | | | | | | | | | | 0.02909 | 16 | | |

[a]AIC = Annualized initial cost  
M = Maintenance cost  
F = Fuel and lubrication cost  
TAC = Total annualized cost

## TABLE 8-12. COST OF ALTERNATIVE NO$_x$ CONTROLS FOR DUAL-FUEL ENGINES FOR ELECTRICAL GENERATION

| NO$_x$ Control % Reduction | | Cost Component[a] | Retard (R) | | Mnfd Cooling (M) | | External EGR | | Derate (D) | | Air-to-Fuel (A) | | R+M | | R+A | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | $/hp-hr | $/hp-hr | %Chg | $/hp-hr | %Chg | $/hp-hr | %Chg | $/hp-hr | %Chg | $/hp-hr | %Chg | $/hp-hr | %Chg | $/hp-hr | %Chg |
| 20% | Low | AIC | 0.00375 | 0 | 0.00381 | 1.5 | 0.00394 | 5 | 0.00426 | 14 | 0.00375 | 0 | | | | |
| | | M | 0.004 | 33 | 0.0035 | 17 | 0.0054 | 80 | 0.00341 | 14 | 0.0031 | 3 | | | | |
| | | F | 0.02168 | 1 | 0.02168 | 1 | 0.02168 | 1 | 0.02146 | 0 | 0.02146 | 0 | | | | |
| | | TAC | 0.02943 | 4.3 | 0.02899 | 2.8 | 0.03102 | 10 | 0.02973 | 3.3 | 0.02831 | .4 | | | | |
| | High | AIC | 0.00375 | 0 | | | | | 0.005 | 33 | 0.00375 | 0 | | | | |
| | | M | 0.004 | 33 | | | | | 0.004 | 33 | 0.0031 | 3 | | | | |
| | | F | 0.02210 | 3 | | | | | 0.02318 | 8 | 0.02188 | 2 | | | | |
| | | TAC | 0.02985 | 5.8 | | | | | 0.03218 | 14.1 | 0.02873 | 1.8 | | | | |
| 40% | Low | AIC | 0.00375 | 0 | 0.00381 | 1.5 | | | 0.00536 | 43 | 0.00375 | 0 | 0.00381 | 1.5 | | |
| | | M | 0.004 | 33 | 0.0035 | 17 | | | 0.00429 | 43 | 0.0031 | 0 | 0.0035 | 17 | | |
| | | F | 0.02188 | 2 | 0.02168 | 1 | | | 0.02233 | 4 | 0.02188 | 2 | 0.02210 | 3 | | |
| | | TAC | 0.02963 | 5 | 0.02899 | 2.8 | | | 0.03177 | 13.3 | 0.02873 | 1.8 | 0.02941 | 4.3 | | |
| | High | AIC | | | | | | | 0.0075 | 100 | | | | | | |
| | | M | | | | | | | 0.006 | 100 | | | | | | |
| | | F | | | | | | | 0.01159 | 8 | | | | | | |
| | | TAC | | | | | | | 0.03668 | 30 | | | | | | |
| 60% | Low | AIC | 0.00375 | 0 | | | | | 0.0075 | 100 | | | | | 0.00375 | 0 |
| | | M | 0.004 | 33 | | | | | 0.006 | 100 | | | | | 0.0041 | 36 |
| | | F | 0.02188 | 2 | | | | | 0.02404 | 12 | | | | | 0.02168 | 1 |
| | | TAC | 0.02963 | 5 | | | | | 0.03754 | 33 | | | | | 0.02953 | 4.7 |
| | High | AIC | | | | | | | | | | | | | 0.00375 | 0 |
| | | M | | | | | | | | | | | | | 0.0041 | 36 |
| | | F | | | | | | | | | | | | | 0.02110 | 3 |
| | | TAC | | | | | | | | | | | | | 0.02995 | 6.2 |

[a] AIC = Annualized initial cost
M = Maintenance cost
F = Fuel and lubrication cost
TAC = Total annualized cost

## TABLE 8-13. COST OF ALTERNATIVE $NO_x$ CONTROLS FOR NATURAL GAS ENGINES FOR OIL AND GAS TRANSPORT

| $NO_x$ Control % Reduction | | Cost Component[a] | Retard (R) | | Mnfd Cooling (M) | | External EGR | | Derate (D) | | Air-to-Fuel (A) | | R+M+A | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | $/hp-hr | $/hp-hr | %Chg | $/hp-hr | %Chg | $/hp-hr | %Chg | $/hp-hr | %Chg | $/hp-hr | %Chg | $/hp-hr | %Chg |
| 20% | Low | AIC | 0.00375 | 0 | 0.00381 | 1.5 | 0.00394 | 5 | 0.00426 | 14 | 0.00375 | 0 | | |
| | | M | 0.003 | 0 | 0.0031 | 3 | 0.0054 | 80 | 0.00341 | 14 | 0.0031 | 3 | | |
| | | F | 0.01556 | 1 | 0.01540 | 0 | 0.01540 | 0 | 0.01570 | 2 | 0.01540 | 0 | | |
| | | TAC | 0.02231 | .7 | 0.02231 | .7 | 0.02474 | 11.7 | 0.02337 | 5.5 | 0.02225 | .5 | | |
| | High | AIC | 0.00375 | 0 | | | | | 0.00577 | 54 | 0.00375 | 0 | | |
| | | M | 0.003 | 0 | | | | | 0.00462 | 54 | 0.0031 | 3 | | |
| | | F | 0.0160 | 4 | | | | | 0.01632 | 6 | 0.01570 | 2 | | |
| | | TAC | 0.02275 | 2.7 | | | | | 0.02671 | 20.6 | 0.02255 | 1.8 | | |
| 40% | Low | AIC | 0.00375 | 0 | | | | | 0.00426 | 14 | 0.00375 | 0 | 0.00381 | 1.5 |
| | | M | 0.003 | 0 | | | | | 0.00341 | 14 | 0.0031 | 3 | 0.0032 | 6 |
| | | F | 0.01570 | 2 | | | | | 0.01570 | 2 | 0.01570 | 2 | 0.01648 | 7 |
| | | TAC | 0.02245 | 1.4 | | | | | 0.02337 | 5.5 | 0.02255 | 1.8 | 0.02349 | 6 |
| | High | AIC | | | | | | | 0.0075 | 100 | | | | |
| | | M | | | | | | | 0.0060 | 100 | | | | |
| | | F | | | | | | | 0.01910 | 24 | | | | |
| | | TAC | | | | | | | 0.03260 | 47 | | | | |
| 60% | Low | AIC | | | | | | | 0.00469 | 25 | 0.00375 | 0 | 0.00381 | 1.5 |
| | | M | | | | | | | 0.00375 | 25 | 0.0031 | 3 | 0.0032 | 6 |
| | | F | | | | | | | 0.01570 | 2 | 0.01570 | 2 | 0.01648 | 7 |
| | | TAC | | | | | | | 0.02414 | 9 | 0.02255 | 1.8 | 0.02349 | 6 |
| | High | AIC | | | | | | | 0.00625 | 66 | 0.00375 | 0 | | |
| | | M | | | | | | | 0.005 | 66 | 0.0031 | 3 | | |
| | | F | | | | | | | 0.01878 | 22 | 0.01638 | 5 | | |
| | | TAC | | | | | | | 0.03003 | 36 | 0.02303 | 4.0 | | |

[a] AIC = Annualized initial cost
M = Maintenance cost
F = Fuel and lubrication cost
TAC = Total annualized cost

# TABLE 8-14. COST OF ALTERNATIVE $NO_x$ CONTROLS FOR NATURAL GAS ENGINES FOR OIL AND GAS PRODUCTION

| $NO_x$ Control % Reduction | | Cost Component[a] $/hp-hr | Retard (R) $/hp-hr | %Chg | Mnfd Cooling (M) $/hp-hr | %Chg | External EGR $/hp-hr | %Chg | Derate (D) $/hp-hr | %Chg | Air-to-Fuel (A) $/hp-hr | %Chg | R+M+A $/hp-hr | %Chg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 20% | Low | AIC | 0.00125 | 0 | 0.00127 | 1.5 | 0.00131 | 5 | 0.00142 | 14 | 0.00125 | 0 | | |
| | | M | 0.003 | 0 | 0.0031 | 3 | 0.0054 | 80 | 0.00341 | 14 | 0.0031 | 3 | | |
| | | F | 0.01778 | 1 | 0.01760 | 0 | 0.01760 | 0 | 0.01812 | 3 | 0.01760 | 0 | | |
| | | TAC | 0.02203 | .8 | 0.02197 | .5 | 0.02431 | 11.3 | 0.02295 | 5 | 0.02195 | .5 | | |
| | High | AIC | 0.00125 | 0 | | | | | | | 0.00125 | 0 | | |
| | | M | 0.003 | 0 | | | | | | | 0.0031 | 3 | | |
| | | F | 0.01830 | 4 | | | | | | | 0.01796 | 2 | | |
| | | TAC | 0.02255 | 3.2 | | | | | | | 0.02231 | 2.1 | | |
| 40% | Low | AIC | 0.00125 | 0 | | | | | 0.00156 | 25 | 0.00125 | 0 | 0.00127 | 1.5 |
| | | M | 0.003 | 0 | | | | | 0.00375 | 25 | 0.0031 | 3 | 0.0032 | 6 |
| | | F | 0.01796 | 2 | | | | | 0.01796 | 2 | 0.01796 | 2 | 0.01884 | 7 |
| | | TAC | 0.02221 | 1.6 | | | | | 0.02327 | 6.5 | 0.02231 | 2.1 | 0.02331 | 6.7 |
| | High | AIC | | | | | | | 0.00156 | 25 | | | | |
| | | M | | | | | | | 0.00375 | 25 | | | | |
| | | F | | | | | | | 0.01830 | 4 | | | | |
| | | TAC | | | | | | | 0.02361 | 8.1 | | | | |
| 60% | Low | AIC | | | | | | | 0.00167 | 33 | 0.00125 | 0 | 0.00127 | 1.5 |
| | | M | | | | | | | 0.004 | 33 | 0.0031 | 3 | 0.0032 | 6 |
| | | F | | | | | | | 0.01830 | 4 | 0.01796 | 2 | 0.01884 | 7 |
| | | TAC | | | | | | | 0.02397 | 9.7 | 0.02231 | 2.1 | 0.02331 | 6.7 |
| | High | AIC | | | | | | | 0.00167 | 33 | 0.00125 | 0 | | |
| | | M | | | | | | | 0.004 | 33 | 0.0031 | 3 | | |
| | | F | | | | | | | 0.01866 | 6 | 0.01848 | 5 | | |
| | | TAC | | | | | | | 0.02433 | 11.4 | 0.02283 | 4.5 | | |

[a] AIC = Annualized initial cost
M = Maintenance cost
F = Fuel and lubrication cost
TAC = Total annualized cost

## TABLE 8-15. SUMMARY OF LARGE-BORE ENGINE $NO_x$ CONTROL COST PENALTIES (AS A PERCENTAGE OF UNCONTROLLED TESTS) BY END USE

| FUEL/APPLICATION | 20% REDUCTION | | | 40% REDUCTION | | | 60% REDUCTION | | |
|---|---|---|---|---|---|---|---|---|---|
| | CONTROL[a] | AVERAGE | RANGE | CONTROL[a] | AVERAGE | RANGE | CONTROL[a] | AVERAGE | RANGE |
| Diesel/Electric Generation | R,D,E, A,RM, RMA | 7 | 0-31 | R,D, RA, AM | 13 | 3-40 | R,RA | 16 | 14-18 |
| | Excl. D | 4 | 0-8 | Excl. D | 4 | 3-6 | | | |
| | D only | | 9-31 | D only | | 37-40 | | | |
| Dual-Fuel Electric Generation | R,M,E, D,A | 6 | 1-14 | R,M,D,A RM | 7 | 2-30 | R,D,RA | 14 | 5-33 |
| | Excl. D,E | 3 | 1-6 | Excl. D | 4 | 2-5 | Excl. D | 5 | 5-6 |
| | D only | | 3-14 | D only | | 13-30 | D only | | 33 |
| Natural Gas/Oil and Gas Transmission | R,M,E D,A | 6 | 1-21 | R,D,A RMA | 9 | 1-47 | D,A, RMA | 10 | 2-36 |
| | Excl. D,E | 1 | 1-3 | Excl. D | 3 | 1-6 | Excl. D | 4 | 2-6 |
| | D only | | 6-21 | D Only | | 6-47 | D only | | 9-36 |
| Natural Gas/Oil and Gas Production | R,M E,D,A | 4 | 1-11 | R,D,A RMA | 4 | 2-8 | D,A RMA | 4 | 2-11 |
| | Excl. D,E | 1 | 1-3 | Excl. D | 4 | 2-7 | Excl. D | 5 | 2-7 |
| | D only | | 5 | D only | | 7-8 | D only | | 10-11 |

## TABLE 8-16. RANGE OF COST PENALTIES (AS A PERCENTAGE OF UNCONTROLLED COSTS) FOR MANUFACTURERS' AVERAGE MODELS

| TYPE OF ENGINE | INDUSTRY AVERAGE REDUCTION[a] | MANUFACTURER | | | | | | | | MAXIMUM RANGE |
|---|---|---|---|---|---|---|---|---|---|---|
| | | Colt | DeLaval | Ing-Rand | Cooper | Superior | Waukesha | Caterpillar | GM/EMD | |
| Gas | 0% | 0% | 1-3% | 1-3% | 0% | 1-3% | 0% | 1-3% | | 0-3% |
| | 20% | 0% | 2-7% | 2-7% | 1-3% | 2-7% | 1-3% | 1-3% | | 0-7% |
| | 40% | 1-3% | 2-7% | 2-7% | 2-7% | 2-7% | 2-7% | 2-7% | | 1-7% |
| | 60% | 2-7% | 2-7% | 2-7% | | 2-7% | 2-7% | 2-7% | | 2-7% |
| Dual-Fuel | 0% | 1-6% | 0% | NA | 1-6% | 1-6% | NA | NA | NA | 0-6% |
| | 20% | 2-5% | 0% | | 1-6% | 1-6% | | | | 0-6% |
| | 40% | 5-6% | 0% | | 2-5% | 2-5% | | | | 0-6% |
| | 60% | 5-6% | 1-6% | | 5-6% | 5-6% | | | | 1-6% |
| Diesel | 0% | 0% | 0% | NA | 0-8% | 0% | NA | NA | 0-8% | 0-8% |
| | 20% | 0-8% | 0-8% | | 3-6% | 0% | | | 3-6% | 0-8% |
| | 40% | 3-6% | 0-8% | | 3-6% | 0-8% | | | 14-18% | 0-18% |
| | 60% | 14-18% | 14-18% | | 14-18% | 3-6% | | | 14-18% | 3-18% |

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[a] Since average uncontrolled $NO_x$ emission levels for each manufacturer vary, cost penalty ranges for a particular $NO_x$ reduction level and fuel type will vary among manufacturers. Moreover, costs are incurred by some manufacturers at 0% reduction since average uncontrolled $NO_x$ emissions from their engines are greater than industry average uncontrolled $NO_x$ emissions.

NA = Not applicable.

cost penalty ranges among manufacturers at the 40- and 60-percent reduction levels; however, the variation within a cost penalty range is large enough (e.g., 2 to 7 percent) for cost differentials to exist among manufacturers. That is, two manufacturers could be in the 2- to 7-percent cost penalty range and one could incur a 2-percent penalty while the other incurred a 7-percent penalty. (The potential impacts of these differentials are discussed in Section 8.4.1.)

This observation also holds for the dual-fuel and diesel categories. The results for the diesel category indicate that Superior Division of Cooper has a cost advantage. However, Superior diesels are smaller, and in general, serve smaller power applications than Colt, DeLaval, or larger Cooper engines. Futhermore, the data are based on results from only one diesel engine model; therefore, the magnitude of this penalty may not be representative of penalties for all of these engines at this level of reduction. The data from Table 8-16 is analyzed in detail in Section 8.4.

### 8.2.2 Engine Manufacturers

Manufacturers of stationary reciprocating IC engines will incur additional costs due to the proposed standards of performance. As discussed in Section 6.3, these costs are a result of one or more of the following activities that may be needed to manufacture engines which meet standards of performance:

- Extended testing to verify the effectiveness of a particular control approach
- Development and application of $NO_x$ controls to existing engine designs

- Engineering, tooling, and pattern costs for the redesign of an engine family

Costs related to these actions have been estimated for the control techniques summarized in Section 6.3 (technically viable approaches to meet proposed standards of performance) and are shown on Table 8-17(60,61). These estimates are representative of the costs that manufacturers would incur to adapt each of the $NO_x$ control systems to an engine family (i.e., group of engines with same air and fuel charging system and combustion chamber geometry). These figures include costs to test engines for durability and to retool their production facilities where necessary, but do not include costs to purchase or manufacture components placed on the engine. These latter costs are included in the user-oriented cost analyses of Section 8.2.1, which considered primarily additional hardware costs in the initial price. The table also gives estimated times to implement the various control technologies. It should be noted that these costs will double if a manufacturer is required to meet emission standards for two types of fuel (e.g., diesel and dual-fuel).

In general, manufacturers believe their present overhead budgets are sufficient for the development of the controls shown on Table 8-17, with the possible exception of EGR and combustion chamber modifications which will require considerably more development over a longer time. As shown in Section 4.3.1, all of the manufacturers have established baseline emissions for most of their engines. They believe that controls such as retard, air-to-fuel, manifold air cooling, derating, and combinations of these approaches would be relatively simple to implement, although some development time would be required to optimize a particular approach and

## TABLE 8-17. ESTIMATED COSTS FOR ENGINE MANUFACTURERS TO DEVELOP $NO_x$ CONTROLS, (BASED ON REFERENCES 60, 61)

| CONTROL | DEVELOPMENT TESTS[a] TIME, HR | COST, $ | EXTENDED TESTS[b] (DURABILITY) TIME, HR | COST, $ | ESTIMATED TOTAL COST[c], $ | TOTAL DEVELOPMENT TIME[d], MONTHS |
|---|---|---|---|---|---|---|
| Retard, R | 200 | 25,000 | 2,000 | 100,000 | 125,000 | 15 |
| Air-To-Fuel Change, A | 200 | 25,000 | 2,000 | 100,000 | 125,000 | 15 |
| Manifold Air Temperature Reduction, M | 400 | 50,000 | 2,000 | 100,000 | 150,000 | 15 |
| Derate, D | 200 | 25,000 | 0 | 0 | 25,000 | 9 |
| R + M | 300 | 40,000 | 2,000 | 100,000 | 140,000 | 15 |
| R + M + A | 400 | 50,000 | 2,000 | 100,000 | 150,000 | 15 |
| External Exhaust[e] Gas Recirculation, EGR | 2,500 | 300,000 | 12,000 | 450,000 | 750,000 | 35 |
| Combustion Chamber Redesign[e] | 10,500 | 1,260,000 | 16,000 | 560,000 | 1,820,000 | 69 |

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[a] This is an estimate of exploratory and development time to establish operation and application data.

[b] This estimate assumes that one technician is in attendance full-time during a test of a 2000-hp engine.

[c] This includes research and development costs, durability tests, and retooling costs. Total cost estimates apply to one engine model family.

[d] From Table 6-5.

[e] Estimates for development time and costs include engineering and redesign of engine components.

establish its durability. Since several manufacturers have less experience in emission control research than others, the costs and times shown in Table 8-17 for these techniques have been estimated conservatively, taking into account the variation in emission control experience among manufacturers.

As discussed in Section 4.4, modifications of engine operating conditions, such as retard or air-to-fuel changes, may necessitate modifications of materials used for exhaust values and turbochargers, if exhaust temperatures greater than 1200°F (present material limits) are experienced. However, temperature data submitted with the emission data of Appendix C do not indicate excessive temperature will be encountered for the levels of retard and air-to-fuel changes reported.

Research and development of $NO_x$ emission control techniques that require more extensive development and/or redesign (e.g., EGR or combustion chamber changes) are difficult to quantify. They will be determined to a large extent by a particular manufacturer's experience with emission controls, the uncontrolled emission levels of his engines, and the response of a particular engine to the control technique. Manufacturers have indicated that the time required to incorporate major engine design changes range typically from 3 to 5 years, but may extend to 10 years in some cases(62,63). This time includes initial design, endurance testing (including 1 to 2 years operation in a real application, but under close monitoring by the manufacturer), and tooling-up (9 to 12 months). It is unclear to what extent the costs for these activities would be added to present R&D expenditures or absorbed into the existing budgets, thereby displacing R&D that would have been undertaken in the absence of emission standards.

These issues will be addressed further in Section 8.4 which discusses the economic impact of both users and manufacturers of IC engines subject to standards of performance.

### 8.2.3 Emerging Controls

In this subsection, the costs to control $NO_x$ emissions from large-bore engines using exhaust gas treatment, combustion chamber modifications, or water induction are estimated. None of these systems has been used on a large-bore engine for any length of time; hence they cannot be considered for immediate application. At the same time they are not dependent upon any technological breakthroughs and should be available for use within 6 years, given appropriate priorities in the R&D budgets of the engine manufacturers.

#### Exhaust Gas Treatment: Ammonia/Catalyst System

Even though no engine manufacturer has reported on the application of $NO_x$ reduction catalysts to large-bore engine exhausts, at least one source considers the reduction of $NO_x$ by ammonia injection over a precious metal (e.g., platinum) catalyst as a promising control technique of the future (see Section 4.4.9)(64). An estimate of the cost of this technique is presented in order to provide a comparison with the costs presented in Section 8.2.1 for the other $NO_x$ controls.

The above-mentioned source reports that approximately 2 cubic feet of honeycomb catalyst (platinum-based) would be required for a 1000-hp engine to ensure proper operation of the system. The cost of the catalyst was estimated at $1500/cubic foot (in 1973). Assuming that the engine costs $150/hp and that the cost of the catalyst accounts for about onehalf the cost of the whole system (container, substrate, and catalyst), the capital investment for this control system represents approximately 4 percent of the engine purchase price. By comparison, catalyst systems for passenger automobiles represent approximately 3 to 5 percent of the purchase price of the automobile ($150 catalyst for a $3000 to $5000 automobile). If, however, this cost is expressed as a percent of the automobile's engine cost ($300 to $500), then the cost of the catalyst system is 30 to 50 percent of the initial price.

The amount of ammonia required for an ammonia/catalyst $NO_x$ reduction system will depend on the $NO_x$ emission rate (g/hp-hr). Based on uncontrolled $NO_x$ emission rates of 9 to 22 g/hp-hr, as reported in Chapters 3 and 4, and the cost of $150/ton for the ammonia, the cost impact of injecting ammonia is approximately 5 to 15 percent of the total annual operating costs ($/hp-hr) for natural gas engines (see Table 8-8)[3/]. When this operating cost is combined with the capital cost of the catalytic system discussed above, the total cost increase is about 25 percent. Therefore, in continuous service applications this system is expensive compared to control techniques such as retard or air-to-fuel changes. However, the cost effectiveness ($ per mass $NO_x$ removal) of an ammonia /catalyst $NO_x$ reduction system could be quite low if the system is as effective on the exhaust from an IC engine as it is on the tail gas from nitric acid production plants. Nitrogen oxide reductions of approximately 90 percent have been reported in such cases, but the exhaust contained

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[3/] This cost approximation is based on a stoichiometric ratio of ammonia and nitrogen dioxide. On this basis 0.6 g $NH_3$ will reduce 1 g NO or 0.7 g $NH_3$ will reduce 1 g $NO_2$.

less than 5-percent oxygen(65). Large-bore reciprocating engine exhaust contains considerably more oxygen (e.g., about 15 percent for gas engines); hence $NO_x$ reduction may be less.

It is also important to note that the consumption of ammonia can be expressed as a quantity of fuel since natural gas is generally used to produce ammonia. Assuming a conservative $NO_x$ emission rate of 20 g/hp-hr, and engine heat rate of 7500 Btu/hp-hr, a heating value of 21,800 Btu/lb for natural gas, and a requirement for approximately 900 lbs of gas per ton of ammonia produced(66), then the ammonia necessary for the catalytic reduction has the same effect on the supply of natural gas as a 2-percent increase in fuel consumption. Additional fuel is required to operate the plant which produces the ammonia.

### Combustion Chamber Modifications

Section 4.4 described several chamber designs that have been shown to produce relatively low $NO_x$ emissions from truck-size engines. There seems to be no technological reason why these designs cannot be adapted to larger engines (careful engineering analysis and design would be required to make the transition). Moreover, it was noted in Section 4.4 that a variable throat precombustion chamber design had been adapted to a large-bore engine in a laboratory. Therefore, an estimate of the costs associated with such a change is presented.

It is difficult to estimate the cost of designing a new, major change to an engine. One large-bore manufacturer estimates that a combustion chamber redesign would require 4 to 5-1/2 years to complete depending on the engine design (2- or 4-stroke cycle in this case). This redesign could affect pistons, cylinder heads and liners, injection components, and valves and would require an additional 12-month endurance testing before the final design could be released for production. This manufacturer would not need to expand his facilities for such a redesign program; nevertheless, he estimates the R&D expenditure would be in addition to normal development work and on the order of 1-1/2 times their typical development costs.

The estimate to be presented here comes from private contacts with manufacturers of truck-size engines who have considered the advisability of converting their direct-injection engines to precombustion chambers. They estimate an approximate development cost of $0.5 million, and note that this figure would be higher if their staff had not already developed familiarity with precombustion chambers through previous experiences. The estimate is based also on the experience of Teledyne Continental who reported a $3-million development and retooling cost for a gasoline engine with a new chamber and ignition system design to replace an existing production model(67). This engine is rated at less than 100 horsepower, and several tens of thousands are produced each year (14,700 alone for stationary applications in 1974). Teledyne Continental's expenditure probably represents an upper bound for the costs that a large-bore engine manufacturer might face; his R&D costs might be more, but retooling expenses should be significantly less.

Based on this information, we estimate that it would cost a manufacturer of large-bore engines no more than $2 million to convert his units to precombustion chambers or squish lips. If one assumes that the average manufacturer in this group sells 50 engines a year, that the average rating of these engines is 1200 hp, that the cost of $150/hp applies, and that the manufacturer should be able to recover such an investment in

5 years, then he would have to increase the sales price of his engines by nearly 5 percent.

Presumably fuel consumption in these redesigned engines could also be 5 to 8 percent higher than it is for current models, just as in some precombustion chamber truck-size engines relative to their closest open-chamber equivalents. In that case, the economic impact from a standard that could be met only by such a redesign would be about 5 to 8 percent of the uncontrolled total annual costs for engines used in continuous service, depending on fuel penalty.

Water Induction

Research has indicated that the induction of water into the engine in quantities equivalent to the mass of fuel consumed can be effective in reducing $NO_x$ emissions(68,69). Reductions of approximately 60 percent in $NO_x$ have been demonstrated on several gaseous fueled engines, using water to fuel mass flow ratios (lb water/lb fuel) of nearly one. In general, no increase in fuel consumption occurred (see Chapter 4 and Appendix C). However, this research indicated that serious maintenance and durability problems are associated with water induction, and therefore, this control technique has been omitted from the cost analysis of currently available techniques (Section 8.2.1).

Since this technology, however, could become available in the future after further testing with water treatment, different allowances in the water injectors, exhaust valves, etc., costs are estimated below for this potential technology.

Testing has shown that the water must be deionized prior to induction to remove minerals which would otherwise deposit in the engine (e.g., on the intake and exhaust valves) and adversely affect performance(70).

For a 1000-hp engine, approximately 1 gallon per min- ute would be required to obtain a water/fuel ratio of 1. Deionized water could be supplied in bulk with an appropriately sized storage facility at a price of $15/1000 gallons, excluding the initial price of the storage facility(71). Based on the information supplied in Table 8-8, such a system would cause the total annual cost (including water treatment and disposal, storage, and delivery system) for a 1000-hp gas or dual-fuel engine used in continuous service (8000 hr/yr) to increase about 8 percent, assuming no additional maintenance. Total annual costs, however, would increase approximately 25 percent if one made the reasonable assumption that engine maintenance requirements would be doubled (cleaning flow passages, dewatering or replacing lubricant, etc.). Corresponding increases for a similarly sized diesel-fueled engine would be approximately one-half as much, since diesel fuel costs about 2.5 times more than natural gas.

If the water is deionized at the engine location, the water cost will be approximately $50/1000 gallons for a system that supplies 1 gallon per minute(72). Total annual costs for a 1000-hp engine in continuous service fueled with natural gas would increase approximately 15 percent for no engine maintenance increase and 85 percent assuming engine maintenance costs doubled.

A larger engine, e.g., 6000-hp, would require a proportionately greater water rate to maintain a water/fuel ratio of one. A reverse osmosis water treatment system would be better suited for this higher volume application (≃5 gallons per minute). Based on a $50,000 investment for this system and a raw water cost of $0.5/1000 gallons, the total nnual costs for a 6000-hp gas-fueled engine in continuous duty

(8000 hr/yr) would increase approximately 10 percent assuming no increase in engine maintenance, and 35 percent assuming maintenance charges doubled.

### Summary of Emerging Control Techniques

The costs associated with an exhaust gas treatment system (ammonia) and with combustion chamber modifications are comparable to those for derating and EGR (see Table 8-15). Water induction control costs appear to be generally higher than for most control techniques summarized in Table 8-15, especially if deionized water is not readily available in bulk, and if maintenance is significantly increased.

## 8.2.4 Fuel Pretreatment

### Desulfurization

Sulfur oxides arise from the nearly quantitative combustion of sulfur in the fuel. The fuels traditionally burned in reciprocating engines (i.e., gas and distillate oil) are low in sulfur, nitrogen, and ash. According to a recent survey, over 50 percent of all distillate contains less than 0.3 percent sulfur, and the average nitrogen content is about 0.03 percent(73).

As explained in Section 4.4.13 of this report, large-bore diesel engines are occasionally equipped to burn crude or residual fuel oils. These generally contain higher levels of sulfur than the more commonly used distillate fuels.[4] Therefore, engines fired with the heavier high-sulfur fuels may incur increased control costs above those outlined in Section 8.2.1 for $NO_x$ control alone. The purpose of this section of the report is to determine the cost of possible standards for the user who burns residual oil.

[4] The sulfur content of fuel oils, in percent, is roughly comparable to the pounds of SO generated per million Btu heat input per hour. For example, 0.3 percent S results in about 0.3 pounds $SO_2$ per million Btu.

Currently, desulfurization of the lower grade fuels appears to be the preferred route to compliance with $SO_2$ emission standards.[5/] Desulfurization of residual fuel oils is widely used. Given that the cost of producing low-sulfur residual fuel oil depends on the specific characteristics of the crude oil used, the extent of desulfurization required, and many other variables, it is reasonable to establish a range of incremental desulfurization costs as follows(74):

| Desired Sulfur Level (Percent) | Desulfurization Cost ($/bbl) | Incremental cost[a] ($/bbl) | ($/MBtu)[b] |
|---|---|---|---|
| 1.0 | 1.10 to 1.50 | 0 | 0 |
| 0.8 | 1.25 to 1.80 | +0.15 to 0.30 | 0.04 |
| 0.5 | 1.45 to 2.00 | +0.35 to 0.50 | 0.07 |
| 0.3 | 1.60 to 2.20 | +0.50 to 0.70 | 0.10 |

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[a] Referenced to 1 percent sulfur level

[b] The heat content assumed in converting $/bbl to $/MBtu is 145,000 Btu per gallon of fuel oil. Also, values shown for $/MBtu are average values.

In the spring of 1975, residual oil with about 1-percent sulfur content cost $2.15/MBtu(75). Hence, the cost to comply with a strict standard of 0.3 percent sulfur is equivalent to a 4.6-percent increase in fuel price. Since fuel costs account for about 75 percent of the total annual cost of operating an uncontrolled diesel-fueled engine in continuous duty (see Figure 8-9), this degree of desulfurization would result in

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[5/] Since desulfurization is a purposeful system for fuel cleaning, it complies with the 1977 amendments to Section 111 of the Clean Air Act which requires the use of continuous technological systems of emission reduction.

a 3.5-percent increase in total fuel costs. Comparable increases for 0.5- and 0.8-percent sulfur (over 1 percent) are 3.2 and 1.9 percent, respectively.

## Denitrification

While $SO_2$ in the reciprocating engine exhaust originates exclusively from the fuel, $NO_x$ arises from thermal fixation of nitrogen in the air and from partial oxidation of nitrogen in the fuel. The control methods evaluated in Section 8.2.1 were devised only for thermal $NO_x$ and have been tested with the traditional clean fuels (natural gas and distillate oil), where nearly all the $NO_x$ generated is thermal. The possible use of heavier fuels suggests that fuel nitrogen may someday become a problem.

Thus, the currently available technology for control of $NO_x$ from fuel nitrogen is, like that for control of $SO_x$, modification of the fuel. But whereas the desulfurization of residual oil is practiced deliberately, denitrification is practiced almost inadvertently. Some nitrogen is removed as a byproduct of desulfurization. The fraction removed is consistently less than that of sulfur but is rarely monitored. Nevertheless, nitrogen removal occurs at no cost above that for sulfur removal. The fuel is not sold under a nitrogen specification, and indeed the refiners have resisted (successfully, in times of fuel shortages) such a specification. Moreover, desulfurization competes with other refining processes for the limited supply of hydrogen. New developments in desulfurization catalysts have the incentive of reduced hydrogen consumption, but the newer catalysts remove less nitrogen. For the time being, then, the owner of an affected facility cannot buy low-nitrogen fuel by specification as he buys low-sulfur fuel. Low-nitrogen fuel can be used to help reduce

$NO_x$ emissions but cannot be used in lieu of applying control techniques to the engines themselves to meet the standard because the standard(s) will be based on test data from installations burning the traditional clean fuels.

A standard of performance for $NO_x$ which failed to recognize the special problem of fuel nitrogen would create a bias against the burning of residual fuels. One manufacturer indicated crude or residual oil is used in place of distillate where a long-term commitment for the heavy oil is at a price lower than distillate. This spokesman pointed out that since diesel fuel is regulated but not crude, the price of crude is not necessarily lower than distillate. In fact, crude oil must be significantly lower than No. 2 oil to justify the increased capital and maintenance costs for heavy fuel handling(76). Furthermore, the lack of information regarding fuel-bound nitrogen formation and removal precludes a more comprehensive discussion. Therefore, since the proposed standards for diesel and dual-fuel engines will be based on emissions data obtained when using No. 2 oil, compliance with the standards may also require operation with No. 2 oil.

### 8.2.5 Modified Facilities

As discussed in Chapter 5, a user usually does not make physical or operational changes to an existing engine installation which would increase its $NO_x$ emission rate. However, if he did, he would be required to conform to a standard of performance for new sources. Therefore, we will briefly consider the potential cost impacts of such a change.

Essentially the user contemplating a modification has two alternatives:

1. Make the modification, including use of the most practical available control technology (see Chapter 5) that prevents the emissions from increasing as a result of the modifications
2. Buy a new engine, or alternative power source, which satisfies the new requirement for which the modification was intended. If a new engine were purchased, the appropriate standard of performance would apply.

Normally a user would contemplate a modification for one of the following two reasons:

1. To increase the power output of his engine
2. To install newly designed parts, such as injectors or pistons, in place of the old ones during an overhaul

For example, a user might wish to install a new injector that the manufacturer designed to reduce smoke or fuel consumption. Other changes, as discussed in Chapter 5, could include alterations of the cylinder head, piston, valve or porting configuration, and manifolds.

If these changes resulted in greater $NO_x$ emissions, the user would probably rely on the control techniques described in Chapter 6 to bring the modified engine within standards of performance. The cost impact would depend on the initial emission level and the $NO_x$ emission level after modification. If the level after a modification was in the same range as those from new, uncontrolled units, the cost increases would also be similar to those presented in Section 8.2.1. (Derating is excluded as a possible control strategy since power requirements at existing facilities are fixed.)

At present there is insufficient data to judge the response of older engines to $NO_x$ control techniques. Some older engines are likely to have emissions lower than a standard of performance for new sources, and emissions after modification could also be less. In other cases, this might not be so. Therefore, it is not possible to state quantitatively the economic impact for users who make a modification to their engine and are then required to meet a standard of performance.

When faced with the need to meet a standard as a result of an intended modification, an owner or operator would normally weigh the cost impact of this approach against that of replacing his existing facility with a new one. This new engine would, of course, meet the standard as well as satisfy his new need. The cost of a major overhaul is probably a reasonable estimate of the maximum price of a modification-plus-alteration to bring the engine into compliance with a standard of performance for new sources. For a large-bore, low- or medium-speed reciprocating engine, this cost is typically less than one-third (see Chapter 5) of the purchase price of the engine.[6/] Therefore, to be competitive, an alternative engine or power source would need to approximate this cost, taking into account both initial and operating costs. Thus, it is unlikely that the user would substitute a different engine (to comply with performance standards) rather than apply control technology to his existing engine.

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[6/] A spokesman for one engine manufacturer indicated that practically all the development of design changes for new engines occurred in-house with the manufacturer incurring about 90 percent of the cost for endurance testing of the design change. Thus, end user-evaluation of design changes are rare, and hence, those costs are not included in this estimate.

### 8.2.6 Reconstructed Facilities

In large-bore, lower medium-speed engines the main housing, or structure, is never replaced during the engine's life, but at some time or other almost every component may be renewed. In general, the various parts are replaced at different times, as they wear out or break, but not at the same time (except for parts replaced during routine maintenance and overhaul). As noted in Chapter 5, overhauls are performed routinely throughout an engine's life and, therefore, should not be considered a reconstruction even though substantial portions of the engine are replaced (generally with parts identical to those originally installed). Therefore, since reconstruction within the meaning of 40 CFR 60.15 is not expected to occur, there is no need for estimating the cost impact of standards on reconstructions.

Industry spokesmen have expressed concern that a standard of performance might deter the replacement of older engines with newer, more efficient ones(78). It does not seem likely, however, that the potential cost increases due to a standard would affect that decision. In fact, if one accepts an industry estimate that new engines consume about 75 percent as much fuel as older ones(79), then one can show that it is more cost effective to purchase such a new, efficient engine (i.e., one with a fuel consumption rate of 7500 Btu/hp-hr) for continuous duty than to simply maintain the old one. This is true even if the new engine is burdened with a 5-percent initial and 80-percent maintenance cost increase (e.g., with EGR) to represent the maximum cost penalty expected from controls, if it is assumed that the old engine is completely amortized, and if the same maintenance costs are assigned to the old engine as to an uncontrolled new unit. In fact, the difference in favor of the purchase is about 9 percent on an annualized basis.

## 8.3 OTHER COST CONSIDERATIONS

This section identifies any costs incurred by large-bore reciprocating engine users as a result of environmental regulations other than standards of performance for air pollution. Such regulatory requirements might concern solid waste disposal, water pollution control, or noise control. The purpose of this section is to identify incremental costs imposed by these regulatory requirements that may in some way limit the ability of the user to bear the cost of control techniques presented in Section 8.2.

Stationary reciprocating engines do not cause solid waste disposal or water pollution problems. Used lubricating oil is not considered a solid waste because it is either sold to an oil recycler or burned as fuel in a boiler. The only conceivable source of solid wastes or contaminated water from the operation of these engines might arise from water purification and demineralization for a water induction control system. These wastes, however, are currently hypothetical since such systems do not now exist; they may be used in the future if water induction is developed as an emission control technique. Treatment costs to prepare water for use in engine jacket and aftercooler cooling systems are presently accounted for in maintenance charges for these engines, and hence, have been included in the analysis presented in Section 8.2.1.

Similarly, no noise regulations are presently in effect specifically for large-bore reciprocating engines. These engines are typically installed in remote locations (e.g., gas pipelines) or separate buildings (e.g., electric utilities and standby service) where remote control of the engine or process reduces noise exposure. In addition, mufflers are used to reduce noise. However, one manufacturer of large-bore engines has reported that complex noise control systems could be cost-prohibitive to an industry whose manufacturers only sell a few hundred units per year and who typically have limited staff and facilities for noise control research (80). This source reports, nevertheless, that manufacturers are beginning to incorporate noise control into their research and design programs.

End-users, on the other hand, have been faced with Occupational Safety and Health Act (OSHA) regulations limiting worker exposure to noise levels and EPA regulations for the protection of communities from annoyance. One source reports that noise control for engines in the 800- to 2000-hp range can cost as much as $10,000 to $30,000. (81) The level of effort to meet OSHA requirements is often unclear. For example, simple, relatively cheap devices such as earmuffs and earplugs can be used as a last resort when other methods of noise control prove technically unfeasible. However, the definition of technical feasibility is uncertain, and depends on the circumstances of each particular application.

In conclusion, there are no other regulatory requirements that, at present, will limit stationary reciprocating engine user's ability to absorb incremental costs as a result of standards of performance for air pollution.

## 8.4 ECONOMIC IMPACT

This section analyzes the economic impacts of alternative standards of performance for $NO_x$ emissions from large stationary reciprocating internal combustion engines on engine manufacturers (Section 8.4.1), gas and electricity prices (Section 8.4.2), employment (Section 8.4.3), and foreign trade (Section 8.4.4). These impacts have been evaluated for alternative standards of 20, 40, and 60 percent below sales-weighted average emissions of uncontrolled gas, dual-fuel, and diesel engines. The extent of the impact for each alternative, assuming retard, air-to-fuel change, or manifold temperature reduction techniques are used to achieve compliance, is summarized in Section 8.4.5 and in Table 8-18.[7] The following is a brief description of the results of the economic impact analysis.

The capital budget requirements for testing engine models are an estimated $5 million for a 60 percent alternative. These expenditures will be made over a two-year period and could be financed internally by engine manufacturers from profits on internal combustion engine sales. No firm is expected to lose more than seven percent of its sales to competitors. Gas turbine sales will not make additional inroads into sales of reciprocating engines with the possible exception of diesel engines used for electricity generation. The total U.S. electric bill would increase by 0.3 percent when controls are applied to all engines. This level would not be reached until all engines are replaced (full phase-in would take about 30 years). Localities using internal combustion engines exclusively to generate

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[7] Other techniques-derate, combustion chamber modification, and exhaust gas recirculation -- are treated separately in this section. This is done since there is little likelihood they would be employed to meet these alternative standards. In addition, their wide range of possible penalties and applications preclude meaningful analysis.

TABLE 8-18. SUMMARY OF ECONOMIC IMPACTS

| Impact | Alternative $NO_x$ Standards | | |
|---|---|---|---|
| | 60% | 40% | 20% |
| Impact on Manufacturers (Section 8.4.1) | | | |
| Capital Budget Requirements | $5 million over two years; able to generate internally from profits. | $4.5 million | $4.1 million |
| Intra-Industry Competition<br>5% @ 60<br>6% @ 20-40 | Maximum sales loss unlikely to exceed 5% of ICE sales for any firm. | 6% max. loss | 6% max. loss |
| Competition from Gas Turbines | No sales losses likely, except for diesel engines (where data are sketchy). | Possible sales loss for one diesel mfg. | No losses. |
| Impact on Product Prices and Users (Section 8.4.2) | | | |
| Electricity Prices | U.S. electric bill up 0.3% after full phase-in (in localities using only ICEs) could face a 9% max. increase. | U.S. bill up 0.1%; localities up 3% max. | U.S. bill up 0.1%. Localities up 3% max. |
| Gas Prices | Delivered gas prices up 0.4% after full phase-in (in 30 years). | Increase of 0.3% | Increase of 0.1%. |
| Impact on Employment (Section 8.4.3) | Little impact expected. | Same. | Same. |
| Impact on Foreign Trade (Section 8.4.4) | Engine sales unaffected. U.S. oil imports up 0.6%, all imports up 0.1% after full phase-in. | U.S. oil imports up 0.4%. | U.S. oil imports up 0.2%. |

electric power, however, could face a maximum increase of nine percent. Delivered gas prices will increase by 0.4 percent when controls are fully implemented. No loss in jobs will take place nationwide. Local changes will be minor because sales shifts among manufacturers will not be large. U.S. oil imports will increase by 0.6 percent when controls are fully phased-in. Total imports of goods and services will increase by slightly more than 0.1 percent.

## 40 Percent Alternative

The impacts would be similar to those above, but somewhat less in the cases of capital budget requirements, product price increases, and import increases. Capital budget requirements would be $4.5 million because of the need to test fewer models; this is $0.4 million less than for a 60 percent standard. ElectroMotive diesel engines would still be vulnerable to gas turbine competition. The increase in the total U.S. electric bill would be 0.1 percent. The maximum increase for localities would be three percent. Gas prices would increase 0.3 percent. Oil imports would rise by 0.4 percent and total imports by slightly less than 0.1 percent.

## 20 Percent Alternative

Capital budget requirements would be $4.1 million. The possibility of any sales losses to turbine manufacturers would be remote. The total U.S. electric bill would increase by 0.1 percent, and the maximum increase for localities would be three percent. Gas prices would increase 0.1 percent. Oil imports would increase by 0.2 percent and total imports by less than 0.1 percent.

Comparing the impacts among the various alternatives -- 60, 40, and 20 percent -- there is no evidence that any of the alternatives would cause an extraordinary impact. The following sections present a detailed discussion of the economic impacts that were considered.

### 8.4.1 Impact on Manufacturers

The most direct economic effect of alternative standards of performance is on the manufacturers of large stationary reciprocating internal combustion engines. This impact involves three areas -- capital budget requirements, intra-industry competition, and competition with gas turbines.

#### 8.4.1.1 Capital Budget Requirements

To implement $NO_x$ reductions for their engines, manufacturers will require capital outlays to develop and test engine control techniques and to maintain production of existing model engines under emissions regulations. The size of these outlays will depend primarily on the number of models each firm would need to test, the extent of further testing required, the fuel prices paid during testing, and whether or not adequate laboratory facilities were in place. The ability to finance the outlays will depend upon the profitability of the engine line and the ease with which the initial costs could be absorbed by the companies' current capital resources.

A precise estimate of the outlays cannot be determined without a detailed evaluation of specific control levels by each company. Certain models have uncontrolled emissions that already meet the 20 and 40 percent alternatives, although not the 60 percent alternative. These are shown in Table 8-19. Models that already meet the alternative levels without controls would not require testing. Other models with high uncontrolled emissions may be such a minor part of a company's business that the company would drop them rather than test them with controls. The data as collected, however, do not reveal the importance of individual models to the companies. Furthermore, it is also possible that certain models would not have to be tested because of the test results gained from other models. The amount of

## TABLE 8-19. THE CUMULATIVE NUMBER OF MODELS CURRENTLY ACHIEVING VARIOUS ALTERNATIVE STANDARDS

| Alternative Met (Reduction from average uncontrolled emissions) | Diesel | Dual Fuel | Gas | Total |
|---|---|---|---|---|
| 60% | 0 | 0 | 0 | 0 |
| 40% | 3 | 1 | 1 | 5 |
| 20% | 6 | 1 | 2 | 9 |
| Uncontrolled | 11 | 5 | 23 | 39 |

Notes: Totals include only engine models for which data are available. This includes 39 out of a total of 49 models.

previous testing and the stringency of the standard may also determine the outlay required.

As discussed in Section 8.2.2, two phases of testing are required by the manufacturer to establish that an engine can meet an emission standard: 1) development and 2) extended durability tests. The tests used as a data base for this study will have met the development needs of manufacturers in many cases. However, where models must be controlled more than 40 percent, additional development tests may be needed. Such tests would cost about $25,000 per model tested. Ingersoll-Rand and Alco may need to establish in-house testing laboratories at an additional cost of $50,000 for test instrumentation for each firm.

After development testing, engine models must undergo extended tests to prove the durability of emissions reductions and operations. These tests would be about three times more costly than the development tests. As noted in Section 8.2.2, the entire testing process can take 15-18 months. Several models can be tested concurrently, though several development tests might be needed before a manufacturer can select the best control technique. Overall, development and extended durability testing would cost about $100,000 per engine model. This estimate is based upon confidential correspondence from two manufacturers; it could overstate the actual costs for the 20- and 40-percent alternative standards (which would need fewer development tests).

The total number of internal combustion engine models produced by each firm and the estimated capital budget requirement for testing to satisfy standards of performance are shown in Table 8-20. The total industry bill, including the cost of establishing new laboratories at two firms, would be approximately $5,000,000 for a 60 percent alternative (requiring testing of up to 49 models). The bill would be about $500,000 less ($4.5 million) for the 40 percent alternative, and about $900,000 less ($4.1 million) for the 20 percent alternative, since fewer models would need testing (see Table 8-19).

The capital test requirements would be regarded as an added expense for the manufacturers. The expense would be measured against the profitability of each engine line. The larger the profits, the smaller the burden of the expense. Manufacturers would either absorb the added expense by reducing profits, pass it on to customers in the form of higher prices, or drop the engine line as an uneconomic part of the business.

## TABLE 8-20. ESTIMATED CAPITAL BUDGET REQUIREMENTS TO MEET $NO_x$ STANDARDS OF PERFORMANCE

| Manufacturer | Number of Models[a] | Capital Budget Requirements for $NO_x$ Standards of Performance[b] |
|---|---|---|
| Colt | 7 | $ 700,000 |
| DeLaval | 5 | 500,000 |
| Caterpillar | 2 | 200,000 |
| Waukesha | 4 | 400,000 |
| ElectroMotive | 2 | 200,000 |
| Cooper Industries (Cooper & Superior) | 24 | 2,400,000 |
| Ingersoll-Rand | 4 | 450,000 |
| Alco | 1 | 150,000 |
| All Firms | 49 | $5,000,000 |

[a] An engine model is defined by a set of fuel, air charging, number of strokes, and displacement per cylinder (bore and stroke) parameters.

[b] Based on an average test cost of $100,000 per engine model. Ingersoll-Rand and Alco will have additional expenses of $50,000 each to establish test laboratories.

A major question is whether the internal combustion engine manufacturers will have the financial resources from which to fund the initial capital requirements. They can be funded either externally by increasing debt or internally by using current capital budgets or allocating funds from the capital budgets of other divisions in the parent company. Price increases could be used in addition to these financing techniques to recover the expenses over a number of years. If manufacturers were to seek to recover the annualized cost of test outlays over a five-year period, on average they would have to raise engine prices only one percent (ranging among manufacturers from 0.4 to 2.1 percent).

Although any combination of the above financing techniques could be used, engine divisions are likely to operate from their own internal resources. Unlike investments in new products or plants, the capital test outlays would represent investments in established lines of business and thus, would entail much lower risk. Moreover, these divisions are small parts of their parent companies, as Table 8-5 in Section 8.1.2 shows. Therefore, raising debt or obtaining funds from other capital budgets would not be difficult for such an investment.

Although the precise capital budgets cannot be determined without a detailed evaluation of a specific control level by each company, as previously mentioned, the prospects for internal financing can be put in perspective by comparing capital budget requirements with the sales and profits of internal combustion engine operations. This can be done in a rough manner by comparing parent companies' profits as a percentage of sales (see Table 8-21) to their internal combustion engine divisions' additional capital budget requirements as a percentage of sales. The internal combustion engine divisions' test requirements as a percentage of sales cannot be shown in order to preserve the confidentiality of sales data disclosed by manufacturers. However, in no case did the percentage exceed five percent of sales or the ratio of the parent company's after-tax profits to sales. This is true even in the case of Colt or ElectroMotive where parent company profits fell below five percent of sales during the 1975-1976 period.

## TABLE 8-21. FINANCIAL RESOURCES

| Consolidated Parent Firm of Engine Manufacturer | Profits as % of Sales 1976 | 1975 |
|---|---|---|
| Ingersoll-Rand | 5.6 | 7.0 |
| General Motors (ElectroMotive) | 6.2 | 3.5 |
| Colt Industries | 4.9 | 5.1 |
| Caterpillar | 7.6 | 8.0 |
| Cooper Industries (Cooper & Superior) | 7.5 | 6.5 |
| Dresser (Waukesha) | 7.0 | 6.2 |
| TransAmerica (DeLaval) | a | a |
| General Electric | 5.9 | 4.3 |
| All Firms[b] | 6.4% | 5.8% |

[a] TransAmerica is primarily a financial corporation whose sales are not easily compared with sales of products of manufacturing corporations.

[b] Average profit margin computed as a simple average for all firms for which data were available.

SOURCE: Securities and Exchange Commission Corporate 10-K reports and annual reports to stockholders.

Providing that the internal combustion engine part of the business has approximately the same profit margins based on sales as the parent companies, this indicates that in the case of each manufacturer, the additional capital budget could be financed with funds generated internally from domestic sales of large stationary reciprocating internal combustion engines. In addition, testing expenses would not be subject to taxes which would otherwise be applied if the firm earned profits of that amount. For example, if

$100,000 of expenses were incurred instead of realizing $100,000 of gross profit subject to the marginal federal corporate tax rate of 48 percent, the corporation would have $52,000 less cash flow and the U.S. Government would receive $48,000 less taxes. In effect, the expenditure would cost the corporation $52,000 rather than the nominal $100,000 if it were absorbed from profits and not passed on through price increases.

In the event standards were set that required exhaust gas recirculation or combustion chamber modification, the capital budget requirements would change. Based on confidential correspondence with manufacturers, it is estimated that test requirements for the industry would double -- to ten million dollars -- and the testing time required would double or triple -- to three to five years. Hence, although cost doubles, the time involved doubles or triples. This means that the annual requirement stays the same or decreases. This capital requirement would compete more for a company's overall resources because it is larger and more extended. If only one manufacturer had to incur this amount, it might mean less resources for normal product development or investment and thus some competitive disadvantage.

### 8.4.1.2 Intra-Industry Competition

Manufacturers would not have significant differential impacts for alternative standards that require spark retard, air-to-fuel changes, or manifold temperature reduction. If derate, exhaust gas recirculation, or combustion chamber modification were required, cost penalties among engines would be disparate and might cause competitive shifts in the sales shares of manufacturers.

To identify whether or not significant changes in sales would take place, the engine penalty data were analyzed in conjunction with confidential sales data for each manufacturer. Worst-case impacts were determined by looking at the maximum variation possible among engine penalties in a particular market. Where the range of penalties shown in Table 8-16 of Section 8.2.1.3 was small, such as in gas engines, one company would not have an advantage over another company. For instance, at the 40 percent alternative for gas engines it is possible that Waukesha could incur a seven percent penalty and Caterpillar a two percent penalty or vice versa, since both of them would incur cost penalties of from two to seven percent.[8] The maximum differential cost penalty for these two competitors in the gas production market is then five percent (seven percent less two percent).

Cross-price elasticities were then considered using the above results. One spokesman estimated that, in one of their highly competitive markets, he would expect a 10 percent increase in price to lead to a 20 percent decrease in sales.[9] Another spokesman, referring to the internal combustion engine market as a whole, estimated that a five percent increase in price would not have any noticeable effect on their sales, but that a 10 percent increase, even industry-wide, would lead to a 10 percent decrease in sales. At markups beyond 10 percent, we have no estimates of the price elasticities, but have assumed that every one percent increase in price would result in a two percent decrease in sales.

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[8] It should be noted that just because the data show that Caterpillar would have to reduce its emissions by an average of 46 percent versus 33 percent for Waukesha, it does not mean that the Waukesha penalty could not fall into the high end of the range and the Caterpillar penalty into the low end of the range.

[9] This assumes sales are in terms of dollars rather than units. If sales were based on units, the impact on dollar sales would be less.

A third factor considered was the importance of each market to the companies' total internal combustion sales. If standby and export markets (which make up almost half of all large stationary reciprocating internal combustion engine sales) are exempt from standards of performance, this would leave manufacturers with a substantial part of their business unaffected. In addition, as described in Chapter 9.3, smaller bore engines will be exempt from proposed standards of performance. Therefore, only a portion of a manufacturers' engine sales will be affected by standards. Thus differential cost penalties arising from standards would lead to only limited impacts on sales shares.

In combining these factors in a hypothetical example, a manufacturer incurring a possible six percent penalty over his competitors would lose 12 percent of his sales in that market (assuming the worst-case cross-price elasticity), which might be 25 percent of his total sales to all markets -- hence, he would lose only three percent of his total sales. (It should be noted that these are large stationary reciprocating internal combustion engine sales, not total engine sales or parent corporation sales which would make this percentage much smaller.)

In addition, one manufacturer indicated that parts and services accounted for over 25 percent of his annual sales.(82) Since standards of performance would not affect the outstanding population of engines, parts, and services, revenues would provide a stabilizing factor for all manufacturers in the short run, though this would lessen over time.

In the following analysis of intra-industry competition for each of the major submarkets, sales losses of more than 10 percent in any market were used to identify significant effects. For most manufacturers, the potential sales loss was considerably less than 10 percent in most of the submarkets they participated. Moreover, these potential losses were constructed from an admittedly extreme set of assumptions concerning the penalty differential, cross-price elasticities, and cost pass-through. The exact percentage of potential sales loss that could occur under these conditions was withheld to protect the confidentiality of the data.

### 8.4.1.2.1 Electricity Generation

There are two markets to discuss in the area of electricity generation. The first is the dual fuel market supplied by Colt, DeLaval and Cooper/Superior. Based on uncontrolled emission data, DeLaval has a distinct advantage over Colt and Cooper/Superior at the 20, 40 and probably 60 percent alternative levels. The maximum differential impact would be six percent, but no manufacturer would lose 10 percent of its internal combustion engine sales at any of the three control levels.

The second market is the diesel fuel market supplied primarily by Colt, DeLaval, Cooper/Superior, and ElectroMotive. Here, a maximum differential impact of 18 percent is possible at the 40 percent level of control with ElectroMotive suffering the disadvantage. Nevertheless, ElectroMotive has substantial mobile engine sales for locomotives which would be unaffected by proposed standards. It is important to note that the 14-18 percent penalties shown in Table 8-16 are based on data from one engine model. Such limited test data do not necessarily reflect the range of penalties for stringent alternatives since not all engines were tested. Moreover, it is possible that some models might be able to attain low levels of emissions only through techniques like derate or combustion chamber modification at much higher penalties. These would be outside the range shown in the table. It is the likelihood of such additional penalties, rather than the data shown in the test sample, that lead to the possibility of significant differential impacts at stringent levels of performance standards. For these reasons, no firm conclusions can be drawn except that even with an 18 percent differential no manufacturer would lose 10 percent of its internal combustion engine sales.

### 8.4.1.2.2 Gas Production

Ingersoll-Rand, Cooper/Superior, Waukesha, Caterpillar, DeLaval and Colt sell internal combustion engines to the gas production market. As previously mentioned, the average cost penalties for manufacturers would range from one to seven percent -- a maximum differential of six percent. At worst, this would mean that the most any manufacturer would lose would be 12 percent of its sales in this market, but no manufacturer would lose 10 percent of its internal combustion engine sales for all markets at any of the three alternative levels.

### 8.4.1.2.3 Gas Transmission

Cooper, DeLaval, Ingersoll-Rand and Waukesha sell internal combustion engines to the gas transmission market. Because the engines are gas-fueled, cost penalties are similar to those in the gas production market, and no manufacturer would lose 10 percent of its internal combustion engine sales at any of the control levels.

### 8.4.1.2.4 Other Markets

All manufacturers have sales to other markets. These include gas-fueled, dual-fueled, and diesel engines. However, the applications involved are diverse and comprise what are actually many different segmented markets. It would not be accurate to characterize differential penalties across such diverse applications. As noted in subsection 8.4.1.2.3, gas-fueled engines have a low differential penalty and therefore, would not involve much change. Dual-fuel sales to other markets are too small to have a significant impact. Only Colt and ElectroMotive sell diesel engines to other markets. Again, it can be expected that the other markets for diesel engines include many diverse applications. For a 20 percent control alternative, the maximum differential penalty would be five percent. For 40 percent, the maximum average penalty differential increases to 15 percent. This still would not result in a 10 percent loss in internal combustion engines sales for either firm, even if their applications overlapped completely. For the 60 percent alternative, the maximum differential is only four percent for the two firms (but is based on limited data).

### 8.4.1.2.5 Standby, Export, and Small Engines

Standby and export sales of large stationary reciprocating internal combustion engines accounted for 44 percent of all sales of large stationary reciprocating internal combustion engines (by horsepower) in the years 1972-1976 for all the manufacturers. In addition, data on sales of stationary reciprocating internal combustion engines below the regulated size limits (see Chapter 9.3) were not available for this analysis. Assuming these applications are exempt from proposed standards of performance, possible percentage loss of sales for each manufacturer is reduced further. By focusing only on a small section of the $NO_x$ emitting stationary reciprocating engine population (which nonetheless emits the bulk of $NO_x$ from installed sources), the proposed standards of performance affect less than half of the total stationary reciprocating internal combustion engine sales of manufacturers.

#### 8.4.1.2.6 Aggregate Impact on Manufacturers

Because of the broad range of most control penalties, it is not always clear whether the engine manufacturers will gain, lose, or stay even in a given market. This is especially true where combustion chamber modification, derate, or exhaust gas recirculation is involved. It is reasonable to assume that the disadvantages some manufacturers may face in one market may be at least partly offset by advantages they gain in another market. This is highly dependent on the manufacturers' product mix and the cross-price elasticities of each market. It has also been shown in the previous subsections that the industry as a whole is buffered by substantial sales on nonregulated internal combustion engines. The market segmentation within major markets and the importance of other application factors would also soften any impacts.

Of the seven manufacturers studied for intra-industry impacts (Caterpillar, Colt, Cooper/Superior, DeLaval, Electromotive Division of General Motors, Ingersoll-Rand, and Waukesha), only Colt and ElectroMotive appeared to have a clear disadvantage in more than one market at certain control levels. Partly because Colt's sales are concentrated in the highly competitive electricity generation equipment market, and partly because their nonregulated sales are not nearly as significant as ElectroMotive's, Colt could potentially suffer the most significant intra-industry impact.

Assuming that Colt, the most vulnerable manufacturer, were to suffer the most extreme differential in each of the markets in which they participate, and assuming the worst possible cross-price elasticities, Colt would suffer a loss in sales of about six percent.[10]

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[10] Across-the-board standards of 20, 40, and 60 percent could result in six, six, and five percent sales losses, respectively, by Colt.

The conclusion that intra-industry impacts would be sustainable and not cause any major dislocations within the indsutry holds for the degree of control for which the test data represent comprehensive ranges. At stringent levels for the standards (e.g., 60% reduction from uncontrolled data), the possibility arises that some models might require expensive control techniques that would widen the ranges and differentials. Only further testing can ascertain how significant this would be. It is also possible that even at the most stringent levels of control, the differential impact might be insignificant.

### 8.4.1.3 Competition From Gas Turbines

To assess the possible inroads turbines might make in the reciprocating engine market as a result of performance standards on new IC engines, market structure, cost, and other factors must be considered. The three major markets for turbines and reciprocating engines (electricity generation, oil and gas production, and oil and gas transport) are segmented into several submarkets in which factors such as size, weight, durability, reliability, vibration, and ability to handle load variations often dictate the choice of engine. Previous experience with the vendor, reputation, service, and familiarity with existing equipment are usually important considerations in the replacement market.

Turbines do not compete with reciprocating engines based on annualized costs alone, due to their higher operating (fuel) costs, at least not in the normal operating range of reciprocating engines which is 6,000 to 8,000 horus per year. As Table 8-12 shows, reciprocating engines controlled to the

60 percent level are less expensive to own than even uncontrolled turbines, with one possible exception in the electricity generation market [11/]

Table 8-22 also shows that for all the other markets, even the maximum penalty which could be imposed on internal combustion engines would not bring the cost of owning internal combustion engines up to that of turbines.

Proposed $NO_x$ NSPS standards for gas turbines used in electricity generation are estimated to increase their cost by about two percent, and for oil and gas transportation and production applications by one to four percent. Consideration of the proposed $NO_x$ NSPS will not affect the conclusions which can be drawn from this table.

TABLE 8-22. TURBINE VERSUS RECIPROCATING INTERNAL COMBUSTION ENGINES BREAKEVEN ANALYSIS

| Application | Fuel | Maximum Reciprocating Engine Cost Penalty (as % of total annualized costs) | | | Breakeven[a] for 6000 to 8000 hr/yr |
|---|---|---|---|---|---|
| | | $NO_x$ Reduction Alternative | | | |
| | | 20% | 40% | 60% | |
| Electricity Generation | dual fuel | 6% | 6% | 6% | 35-39% |
| | diesel | 8% | 18% | 18% | 12-14% |
| Oil and Gas Transportation | gas | 7% | 7% | 7% | 13-14% |
| Oil and Gas Production | gas | 7% | 7% | 7% | 25-30% |

[a] This represents the total annualized cost penalty which would have to be experienced by reciprocating internal combustion engines before they would equal the cost of uncontrolled turbines.

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[11/] New Source Performance Standards were proposed for stationary gas turbines in the 3 October 1977 Federal Register, Volume 42, Number 191.

### 8.4.1.3.1 Electricity Generation

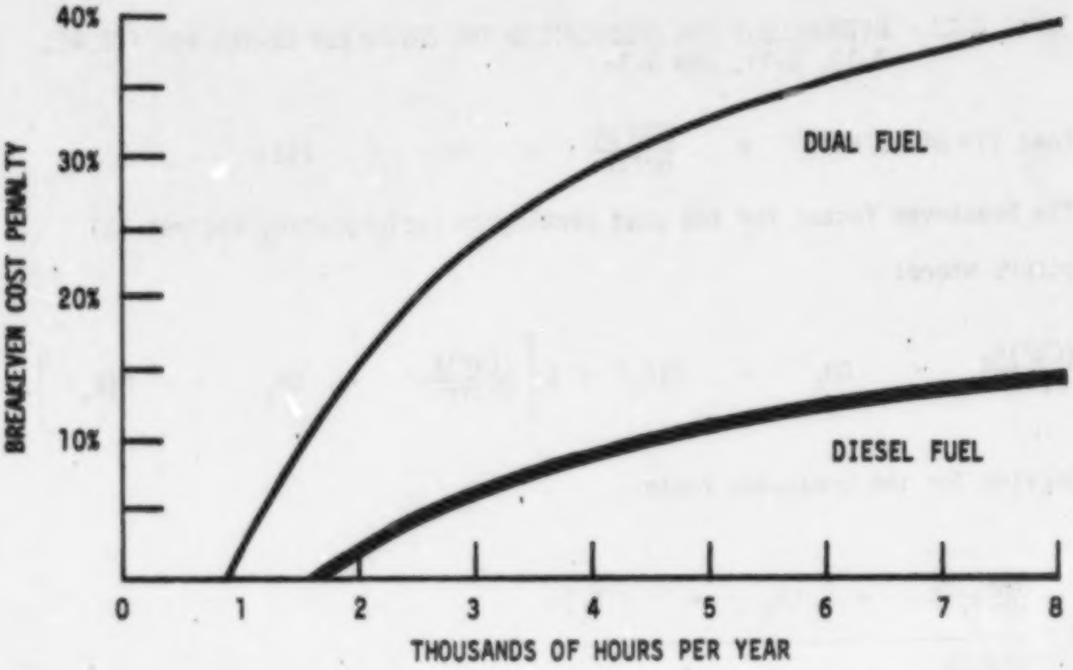
Data shown in Table 8-22 of the cost analysis section indicate that cost penalties of 14-18 percent could be incurred by some diesel engine manufacturers due to controls. One manufacturer, Electromotive, would incur that penalty even at a 40 percent level of $NO_x$ reduction. For all other manufacturers, turbines would still cost more.

Table 8-22 shows that a 12-14 percent cost penalty would bring diesel reciprocating engines used in electricity generation up to the point where they would have no cost advantage over uncontrolled turbines. Since turbines would only incur about a two percent penalty for the proposed $NO_x$ new source performance standard controls,(83) it appears that they would become competitive with diesel reciprocating engines. This conclusion, however, is based on only one data point used to calculate the 14-18 percent diesel engine penalty and, as such, should not be used as a decision criteria, considering that some uncertainty exists at the greater control alternatives (60 and 40 percent) for which there are test data. Furthermore, it is unlikely that turbines would replace diesel engines in plants using banks of smaller reciprocating engines, unless the entire bank were replaced with one turbine.

At the 20 percent alternative, turbines do not compete with the internal combustion engines on a cost basis. As load factors decrease, however, turbines become increasingly competitive due to their lower capital costs as shown in Figure 8-11. The formulas and methodology used to produce this and other graphs illustrating the breakeven points in Table 8-22 are shown separately on Table 8-23.

### 8.4.1.3.2 Oil and Gas Transportation

Although the reciprocating gas engines in gas transportation lose a large portion of their cost advantage over turbines in the high end of the



| | Parameters | | |
|---|---|---|---|
| | | IC Engines | |
| | Turbine-Diesel | Dual Fuel | Diesel |
| CRF (%) | .2 | .2 | .2 |
| K ($/kW) | 120 | 200 | 200 |
| OM ($/kW) | $.05K_t$ | .004 ($/kWh) | |
| P ($/MMBtu) | 2.5 | 2 | 2.5 |
| E ( $\frac{\text{MMBtu}}{\text{kWh}}$ ) | .0132 | .0087 | .0093 |

| Breakeven Factors = X | | |
|---|---|---|
| Hr/Yr | Dual Fuel | Diesel |
| 1,000 | 1.026 | .937 |
| 2,000 | 1.159 | 1.016 |
| 3,000 | 1.238 | 1.060 |
| 4,000 | 1.290 | 1.087 |
| 5,000 | 1.327 | 1.106 |
| 6,000 | 1.354 | 1.120 |
| 7,000 | 1.375 | 1.131 |
| 8,000 | 1.392 | 1.140 |

**Dual Fuel:**

$$X = \frac{\frac{.25(120)}{\text{Hrs/Yr}} + 2.5(.0132)}{\frac{.2(200)}{\text{Hrs/Yr}} + .004 + 2(.0087)}$$

**Diesel:**

$$X = \frac{\frac{.25(120)}{\text{Hrs/Yr}} + 2.5(.0132)}{\frac{.2(200)}{\text{Hrs/Yr}} + .004 + 2.5(.0093)}$$

Figure 8-11. Internal combustion engines vs. turbines in electricity generation applications.

## TABLE 8-23. METHODOLOGY FOR CALCULATING THE BREAKEVEN CURVES FOR FIGURES 8-10, 8-11, AND 8-12

$$\text{Cost (in mills/kWh)} = \frac{(CRF)K}{Hr/Yr} + OM + P(E)$$

The breakeven factor for the cost penalty to reciprocating engines (X) occurs where:

$$\frac{(CRF)K_t}{Hr/Yr} + OM_t + P(E_t) = X\left[\frac{(CRF)K_r}{Hr/Yr} + OM_r + P(E_r)\right]$$

Solving for the breakeven factor:

$$X = \frac{\dfrac{(CRF)K_t}{Hr/Yr} + OM_t + P(E_t)}{\dfrac{(CRF)K_r}{Hr/Yr} + OM_r + P(E_r)}$$

The percentage breakeven cost penalty = 100(X-1), where:

X = breakeven factor

CRF = capital recovery factor (i.e., .2 = 20%)[a]

K = installed cost of engine or turbine ($/kW)

OM = operating and maintenance cost ($/kWh)

P = price of fuel (i.e., distillate $2.50/MMBtu; gas $2.00/MMBtu)

E = heat rate efficiency in MMBtu/kWh

t = turbines

r = large reciprocating internal combustion engines penalty range presented in Table 8-22, they are generally used in different applications, and substitution would not be imminent, even at the breakeven point of total annualized costs.

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[a] Assuming a 20-year accounting life, 10 percent interest rate, and four percent fixed capital charge (includes property taxes, insurance, administration and overhead).

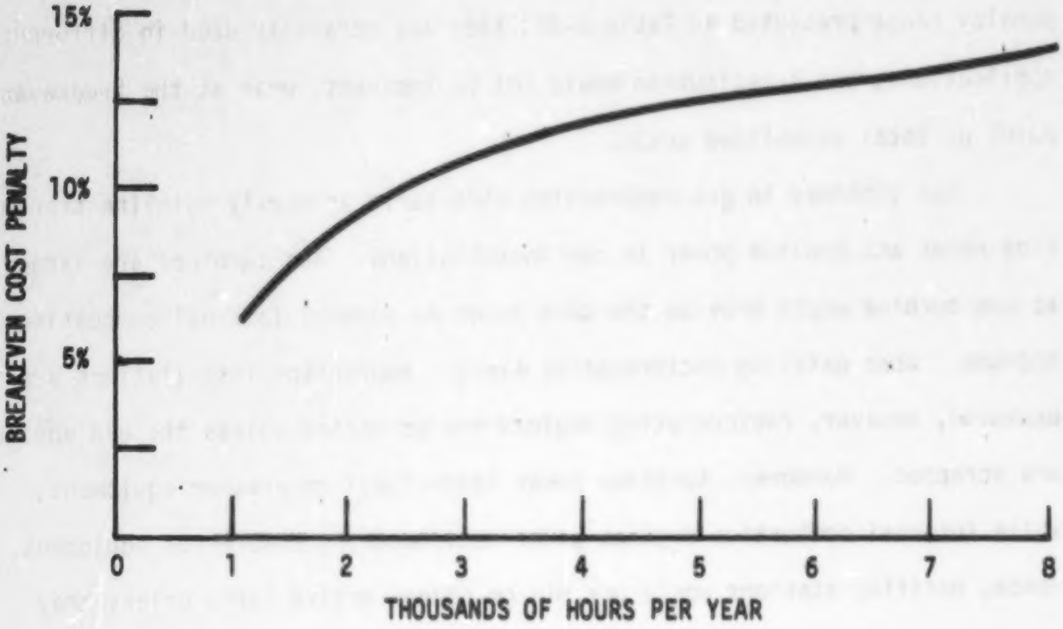
Gas turbines in gas compression uses serve primarily mainline transmission roles and provide power in new installations. Gas turbines are larger, so one turbine would provide the same power as several internal combustion engines. When existing reciprocating engine compression installations are expanded, however, reciprocating engines are purchased unless the old engines are scrapped. Moreover, turbines power centrifugal compressor equipment, while internal combustion engines power reciprocating compressor equipment. Hence, existing stations would not mix or change motive forms unless they also changed their compressor equipment.

Furthermore, turbines are not as well suited for gathering, storage, or pressurization where the flow is highly variable and the discreate power requirements are in the range that internal combustion engines offer. Thus, engines are typically located at the distribution end of pipelines, whereas turbines are located on the main trunk lines.

In addition, high load factors (characteristic of compressor applications) favor the more efficient internal combustion engines, especially as gas prices rise. Figure 8-12 shows that internal combustion engines maintain this advantage, even at load factors as low as 1,000 hours per year.

#### 8.4.1.3.3 Oil and Gas Production

Turbines are clearly uncompetitive on purely a cost basis in this market. They are less efficient and much more expensive than internal combustion engines and are primarily used on offshore rigs where lighter and more portable equipment is a necessity. Uncontrolled turbines used for extraction



**Parameters**

|  | Turbine-Gas | IC Engine-Gas |
|---|---|---|
| CRF (%) | .2 | .2 |
| K ($/kW) | 160 | 200 |
| OM ($/kW) | $.05K_t$ | .004 ($/kWh) |
| P ($/MMBtu) | 2.00 | 2.00 |
| E ($\frac{\text{MMBtu}}{\text{kWh}}$) | .0132 | .0093 |

**Breakeven Factors = X**

| Hrs/Yr | Gas IC Engines |
|---|---|
| 1,000 | 1.061 |
| 2,000 | 1.089 |
| 3,000 | 1.106 |
| 4,000 | 1.117 |
| 5,000 | 1.124 |
| 6,000 | 1.130 |
| 7,000 | 1.134 |
| 8,000 | 1.138 |

$$X = \frac{\frac{.25(160)}{\text{Hrs/Yr}} + 2(.0132)}{\frac{.2(200)}{\text{Hrs/Yr}} + .044 + 2(.0093)} = \frac{\frac{40}{\text{Hrs/Yr}} + .0264}{\frac{40}{\text{Hrs/Yr}} + .0226}$$

Figure 8-12. Internal combustion engines vs. turbines in oil and gas transportation applications.

purposes are at least 25 percent more expensive to own than internal combustion engines, as Figure 8-13 shows.

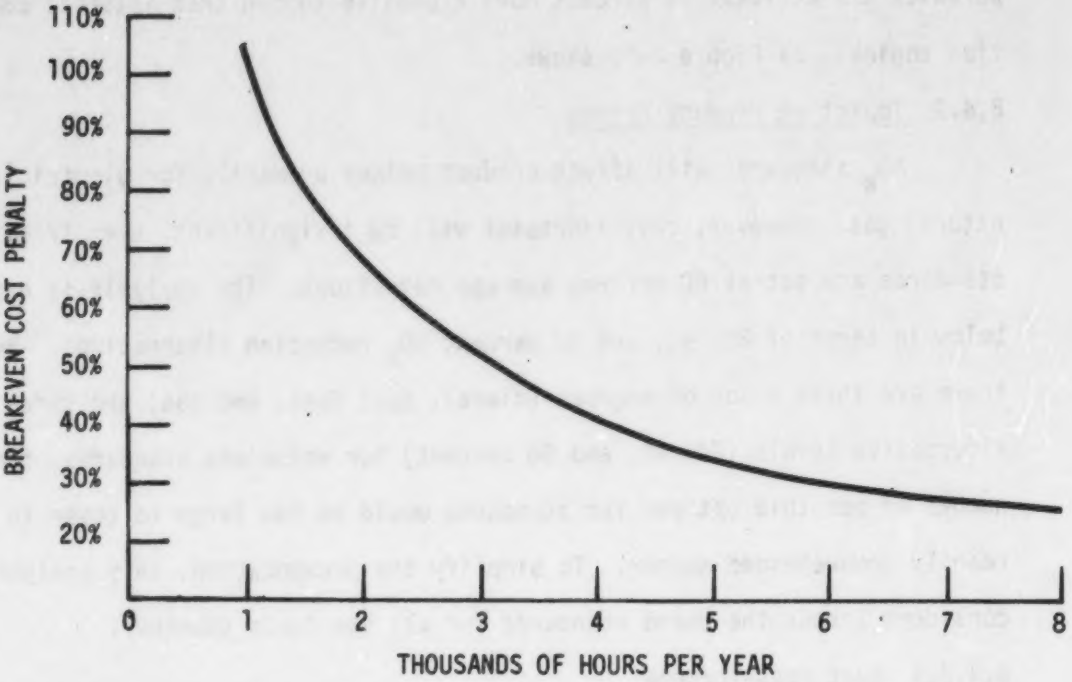
### 8.4.2 Impact on Product Prices

$NO_x$ standards will affect product prices primarily for electricity and natural gas. However, cost increases will be insignificant, even if emissions standards are set at 60 percent average reductions. The analysis is explained below in terms of 20, 40, and 60 percent $NO_x$ reduction alternatives. Because there are three kinds of engines (diesel, dual fuel, and gas) and three alternative levels (20, 40, and 60 percent) for emissions standards, the number of possible options for standards would be too large to cover in a readily comprehended manner. To simplify the presentation, this analysis considers across-the-board standards for all the fuels together.

#### 8.4.2.1 Cost Pass-Through

When the demand for a manufacturer's goods is inelastic (i.e., insensitive to price changes), added costs are likely to be transferred to the consumer. This cost pass-through occurs where the manufacturer's product is viewed as a necessity by the consumer with few, if any, substitutes available. The extent of pass-through, then, depends on many factors in addition to the unique qualities of the product -- the price elasticity of demand for the industry as a whole (determined in part by other demands which are indirectly related) and the cross-price elasticities of the products within the same industry. It is assumed that since manufacturers produce reciprocating engines for essential applications in specialized market, cost increases would be passed through to the consumer.

Section 8.4.1.3 demonstrated that the internal combustion engine industry does not now face competition from gas turbines based on cost factors



**Parameters**

| | Turbine-Gas | IC Engine-Gas |
|---|---|---|
| CFR (%) | .2 | .2 |
| K ($/kW) | 210 | 67 |
| OM ($/kW) | $.05K_t$ | .004 |
| P ($/MMBtu) | 2.00 | 2.00 |
| E ($\frac{\text{MMBtu}}{\text{kWh}}$) | .0136 | .01067 |

**Breakeven Factors = X**

| Hrs/Yr | Gas IC Engines |
|---|---|
| 1,000 | 2.057 |
| 2,000 | 1.668 |
| 3,000 | 1.500 |
| 4,000 | 1.406 |
| 5,000 | 1.345 |
| 6,000 | 1.304 |
| 7,000 | 1.273 |
| 8,000 | 1.250 |

$$X = \frac{\frac{.25(210)}{\text{Hrs/Yr}} + 2(.0136)}{\frac{.2(67)}{\text{Hrs/Yr}} + .004 + 2(.01067)} = \frac{\frac{52.5}{\text{Hrs/Yr}} + .0272}{\frac{13.4}{\text{Hrs/Yr}} + .02534}$$

Figure 8-13. Internal combustion engines vs. turbines in oil and gas production applications.

alone for either of the two main applications--electricity generation and gas compression during production and pipeline transmission--in which $NO_x$ standards will cause cost penalties. In electric generation, it may be recalled, turbines are uneconomic for baseload use and for additions to plants already using banks of reciprocating engines. In gas compression, gas turbines are not suited to gathering applications (because of variable gas flow) and are not easily added to stations already using reciprocating engines.

Furthermore, the industry is segmented into many specialized markets and submarkets, as discussed in Section 8.1.3. Most internal combustion engines have carved out a niche in which they have few, if any, substitutes in the applications for which they are used.

At the same time, demand for engines would not likely fall significantly as a result of higher prices for engines in the range that the test data indicate would take place. The increase in some localities for electric rates would be, at its maximum, somewhat higher--nine percent--but is much less than recent increases from other causes, while past rate increases have not led to much decrease in demand for electricity (see subsection 8.4.2.2). Municipalities have the option of purchasing power from larger utilities, but tend to produce the bulk of their own needs and purchase only small amounts of power. (Frequently, these utilities sell power.) Furthermore, consumers would not consume appreciably less gas because of a 0.4 percent increase in delivered prices (see subsection 8.4.2.3).

With little decrease likely for overall engine demand and little competition from substitutes, manufacturers will likely pass through cost increases to consumers.

### 8.4.2.2 Electricity Generation

Reciprocating internal combustion engines are used to generate electricity on a continuous (baseload) basis for small municipalities. They account for only a small fraction of total electricity generation in the United States. There were 936 internal combustion plants (excluding gas turbines) at year-end 1975.(84) These had a total generating capacity of 5,021 megawatts--equal to one percent of the total U.S. electric generating capacity of 505,772 megawatts.(85) Therefore, the impact on electricity prices is best measured in two dimensions--local and weighted national impacts.

Although localities deriving all of their electricity from internal combustion are rare, and the chances of all of those engines being subject to NSPS are remote at least in the near future, an evaluation of this case was done to provide the maximum possible increase in electricity prices which could be experienced by any consumer. Based on a sample of ten utilities using internal combustion engines to generate at least 90 percent of their electricity sales, engine costs (engine price, maintenance, and fuel expenses) typically account for half the costs of delivered electricity to consumers.(86) Electricity distribution and general overhead costs account for the remainder and would be unaffected by $NO_x$ standards. Therefore, an engine penalty from $NO_x$ standards would be halved when applied to the price that consumers pay for electricity in these localities. Table 8-24 shows the inflationary impact on local electricity prices for various emissions standards.

The maximum impact is nine percent and takes place in the case of diesel engines at a 60 percent alternative (based on data from only one model). Recent sales of diesel and dual fuel engines to the electric

TABLE 8-24. MAXIMUM INFLATIONARY IMPACT ON LOCAL ELECTRICITY PRICES

| Engine type | Alternative $NO_x$ Reductions 20% | 40% | 60% |
|---|---|---|---|
| Diesel | 3% | 3% | 9% |
| Dual Fuel | 2% | 3% | 3% |

generation market indicated that new sales were running approximately 50 percent diesel and 50 percent dual fuel.

In comparison, electric rates have increased far more than other factors. Since 1970 residential, commercial, and industrial electric rates have risen by five percent to 30 percent annually. This is shown in Table 8-25.

TABLE 8-25. HISTORICAL PERSPECTIVE ON ANNUAL INCREASES IN ELECTRICITY PRICES

| Period | Residential 500 kWh | Commercial--30, kW, 6,000 kWh | Industrial--300 kW, 60,000 kWh |
|---|---|---|---|
| 1970 to 1971 | 6.2% | 5.5% | 7.6% |
| 1971 to 1972 | 7.6% | 7.3% | 8.5% |
| 1972 to 1973 | 4.6% | 4.9% | 5.7% |
| 1973 to 1974 | 12.4% | 11.1% | 15.6% |
| 1974 to 1975 | 27.3% | 24.8% | 30.6% |
| 1975 to 1976 | 7.2% | 6.3% | 6.5% |
| Range | 4.6%-27.3% | 4.9%-24.8% | 5.7%-30.6% |

SOURCE: Federal Power Commission, Typical Electric Bills, 1976.

Because internal combustion engines account for just a small fraction of all electricity generated nationwide (with the national electric bill from private utilities alone valued at $44.4 billion in 1976),(87) the $NO_x$ emissions standards on new source reciprocating internal combustion engines would raise the national electric bill by just a fraction of a percent. If the standards had been fully implemented in 1976, the inflationary inpact on the national electric bill would have been just 0.1 percent in the case of 20 or 40 percent alternatives and 0.3 percent in the case of a 60 percent alternative. This is shown in Table 8-26.

TABLE 8-26. MAXIMUM INFLATIONARY IMPACT ON THE NATIONAL ELECTRIC BILL WITH FULL PHASE-IN

| | Alternative $NO_x$ Reductions | | |
|---|---|---|---|
| Impact | 20% | 40% | 60% |
| Increase | 0.1% | 0.1% | 0.3% |

In actuality, new source performance standards are phased-in only gradually. Penalties are not incurred until new controlled engines are purchased, while old uncontrolled engines are retired from service after a 30-year lifetime. Assuming new sales equal to three percent of the existing engine population and retirements also equal to three percent (in effect, a steady population as indicated by recent sales data), after five years, approximately 15 percent of all engines will be controlled and will incur penalties. The inflationary impact of the standards at that time is shown in Table 8-27. It ranges from 0.02 percent for a 20 percent standard to 0.04 percent for a 60 percent standard.

TABLE 8-27. INFLATIONARY IMPACT ON THE NATIONAL ELECTRIC BILL, AFTER FIVE YEARS

| Impact | Alternative $NO_x$ Reductions | | |
|---|---|---|---|
| | 20% | 40% | 60% |
| Increase | 0.02% | 0.02% | 0.04% |

### 8.4.2.3 Gas Production and Transmission

Reciprocating internal combustion engines are used to transport most of the gas consumed in the United States. Internal combustion engine costs account for only six percent of the delivered price of gas ($1.60 per Mcf in 1976),(88) though the percentage is somewhat higher for areas distant from producing states. The average cost penalties for gas engines (which account for nearly all recent sales to the reciprocating pipeline engine market) ranged from two percent at a 20 percent alternative to six percent at a 60 percent alternative. Table 8-28 shows the impact of these penalties on delivered gas prices.

TABLE 8-28. INFLATIONARY IMPACT ON DELIVERED GAS PRICES, AFTER FULL PHASE-IN

| | Alternative $NO_x$ Reductions | | |
|---|---|---|---|
| Impact | 20% | 40% | 60% |
| Increase | 0.1% | 0.3% | 0.4% |

The largest inflationary impact, at a 60 percent alternative, would involve a price increase of just 0.4 percent. (This was calculated as if all engines in use in 1976 had been controlled and incurred penalties.) By comparison, the national average delivered price of gas has risen by 103 percent in recent years--from $.79 per Mcf in 1973 to $1.60 per Mcf in 1976--(89) from other causes.

The increase in gas prices would not reach the above levels until all engines were controlled. With sales and retirement rates equal to those in the electric generation market (i.e., three percent annually), in five years controls would cover 15 percent of all engines. The inflationary impact at that time would be a 0.06 percent increase in the case of the most stringent standard (60 percent). This is shown in Table 8-29.

TABLE 8-29. INFLATIONARY IMPACT ON DELIVERED GAS PRICES, AFTER FIVE YEARS

| | Alternative $NO_x$ Reductions | | |
|---|---|---|---|
| Impact | 20% | 40% | 60% |
| Increase | 0.02% | 0.04% | 0.06% |

### 8.4.2.4 Impacts Over Five Years

Users of internal combustion engines will have to lay out additional capital expenditures to purchase more expensive engines (the engine purchase price component of the cost penalty from $NO_x$ controls). In the case of internal combustion engines, however, the capital cost penalty is small. Most of the penalty comes from higher fuel or maintenance costs. A two percent engine price penalty can be expected on average for all alternative standards.[12/] With annual industry domestic non-standby internal combustion engine sales of $96 million (projected to remain roughly constant), the additional capital cost for users would equal $1.9 million per year -- a total of $9.6 million on a cumulative basis over the first five years.

Total costs in the fifth year (including amortized capital costs, maintenance costs, and fuel costs) would range from $16 million for a 20 percent alternative to $26 million for 40 percent and $45 million for 60 percent.

In dollar terms, the impact of the standards is shown in Table 8-30 for all markets in the fifth year after the standards are implemented.

TABLE 8-30. COSTS OF VARIOUS ALTERNATIVE STANDARDS IN THE FIFTH YEAR (IN MILLION DOLLARS)

| Application | Alternative $NO_x$ Reductions 20% | 40% | 60% |
|---|---|---|---|
| Gas Production & Transmission | 5.9 | 13.6 | 17.8 |
| Electric Generation | 7.9 | 8.9 | 19.9 |
| Other Applications | 2.2 | 3.4 | 7.1 |
| All Applications | 16.0 | 25.9 | 44.8 |

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[12/] Two percent reflects a one percent increase from pass-through of test costs and an average of one percent increase for use of manifold temperature reduction in some cases. (Manifold temperatures reduction would incur a price increase of two percent where used, but would probably be used less frequently than other techniques.)

### 8.4.3 Impact on Employment

Since $NO_x$ standards will not cause significant changes in manufacturers' sales if retard, manifold temperature reduction, and air-to-fuel changes only are used, they will not cause significant impacts on employment.

Nationally, sales shifts among manufacturers will tend to balance out with no decrease in aggregate sales or employment in the industry. As noted in Section 8.4.4 (below), exports and imports of internal combustion engines will not likely experience any changes because of $NO_x$ standards on engines used in the U.S. Therefore, few jobs would be lost to foreign firms. Moreover, because sales changes of greater than 10 percent are not expected (and most would be much less), the extent of local shifts will also be minor.

### 8.4.4 Impact on Foreign Trade

The foreign trade balance will not be significantly affected by $NO_x$ standards. The amount of imported engines is not expected to change. Most internal combustion engine imports have been engines that are smaller than the minimum size to be controlled by the proposed standards. Department of Commerce data showed that the average value of imported diesel engines in 1973 was $889 per unit.(90) This is far below the likely cost for a diesel engine of the size that would be controlled by the proposed new source performance standards since a typical 1,000 horsepower diesel engine would cost about $150,000. In addition, imported engines would have to meet $NO_x$ emissions standards. Because foreign firms would have smaller U.S. sales volumes over which to spread capital test requirements for $NO_x$ reductions, the $NO_x$ standards would actually tend to create a barrier to imports.

Proposed $NO_x$ standards do not apply to engine exports. Control techniques like retard, air-to-fuel changes, manifold temperature reduction, and derate do not involve changes in mass production items. There would be no loss in scale economies to manufacturers if they had to produce controlled engines for domestic sales and uncontrolled engines for export sales. Moreover, even techniques like combustion chamber modification and exhaust gas recirculation would have little impact on exports, because of the specialty nature of the large engine industry, where even without controls, each engine is typically tailored to a specific customer's needs rather than just mass produced.

Fuel imports would be increased marginally by the fuel penalties involved in meeting $NO_x$ standards. In the fifth year after standards take effect, 15 percent of all engines will be controlled. The additional fuel requirement (to be met by additional imports of oil) in that year would be 1.0 million barrels of oil for a 20 percent alternative, 1.5 million barrels for 40 percent, or 2.4 million barrels for 60 percent. This is shown in Table 8-31.

TABLE 8-31. ADDITIONAL FUEL NEEDS, IN THE FIFTH YEAR (MILLION BARRELS)

| Engine Type | Alternative $NO_x$ Reductions 20% | 40% | 60% |
|---|---|---|---|
| Gas | 0.6 | 1.0 | 1.0 |
| Diesel | 0.3 | 0.4 | 1.4 |
| Dual Fuel | 0.1 | 0.2 | 0.1 |
| All Engines | 1.0 | 1.5 | 2.4 |

Measured against 1976 oil imports of 2,850 million barrels,(91) the increase in oil imports would be 0.04 percent for a 20 percent alternative, 0.05 percent for 40 percent, and 0.09 percent for 60 percent. Measured against 1976 imports of all goods and services of $160 billion and valued at an average of $12.10 per barrel,(92) the increase in U.S. imports would be 0.008 percent for a 20 percent alternative 0.011 percent for 40 percent, and 0.018 percent for 60 percent.

### 8.4.5 Summary of Economic Impact Analysis

With minor exceptions, there appears to be little difference among the impacts of a 60 percent, 40 percent, or 20 percent alternative. This is a qualified judgment based on the assumption that derating, exhaust gas recirculation, or combustion chamber modification will not be necessary in order to meet these alternatives.

Although aggregate price impacts on consumers vary directly with the level of control, even a 60 percent alternative would raise gas and electricity prices less than half a percent. On the local level, however, it is theoretically possible for a locality entirely dependent on new reciprocating internal combustion engines for generating its electricity to experience a nine percent increase in electricity prices at the 60 percent alternative. This is three times more than the increases for the 40 or 20 percent alternative. Such localities, however, would represent an extremely small part of the overall population.

Manufacturers would face only limited impacts. Capital test requirements would be within their ability to finance internally from profits, while the costs could be recovered through a one percent average price increase over a five-year period. Despite variations in cost penalties, no firm is likely to lose more than five to six percent of its sales (in all of their markets). Gas turbines may make inroads into certain manufacturers' sales for diesel engines only at a 40 or 60 percent alternative, but their cost advantage in the normal operating range of 6,000 to 8,000 hours per year would be slight. Employment, like sales, would experience little change. Imports and exports of internal combustion engines would also face little change. Oil imports would increase by only a fraction of a percent.

Several additional points can be noted. In summary, the full impact of the standards would not be markedly different for the 20, 40 and 60 percent alternatives. Second, based on conservative analytic techniques, the only possible impact of notable magnitude would be the rise in electric prices for isolated localities using all internal combustion engines. Third, full attainment of the impacts on users will not be realized until all engines are replaced by controlled engines incurring penalties -- a process that will take 30 years to complete.

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# 9. RATIONALE

## 9.1 SELECTION OF SOURCE FOR CONTROL

Previous investigators have concluded that stationary internal combustion (IC) engines are major contributors to nationwide emissions.(1,2,3) In particular, stationary IC engines are sources of $NO_x$, hydrocarbons (HC), particulates, sulfur dioxide ($SO_x$), and carbon monoxide (CO) emissions. $NO_x$ emissions from IC engines, however, are of more concern than emissions of these other pollutants for two reasons. First, $NO_x$ is the primary pollutant emitted by stationary engines. Second, EPA has assigned a high priority to the development of standards of performance limiting $NO_x$ emissions. Assuming existing levels of emission controls, national $NO_x$ emissions from stationary sources are projected to increase by more than 40 percent in the 1975-to-1990 period. Applying best technology to all sources would reduce this increase but would not prevent it from occurring. This unavoidable increase in $NO_x$ emissions is attributable largely to the fact that current $NO_x$ emission control techniques are based on combustion redesign. In addition, few $NO_x$ emission control techniques can achieve large (i.e., in the range of 90 percent) reductions in $NO_x$ emissions. Consequently, EPA has assigned a high priority to the development of standards of performance for major $NO_x$ emission sources wherever significant reductions in $NO_x$ can be achieved. Studies have shown that IC engines are significant contributors to total U.S. $NO_x$ emissions from stationary sources. Figure 9-1(4) shows that internal combustion engines account for 16.4 percent of all stationary source $NO_x$ emissions,

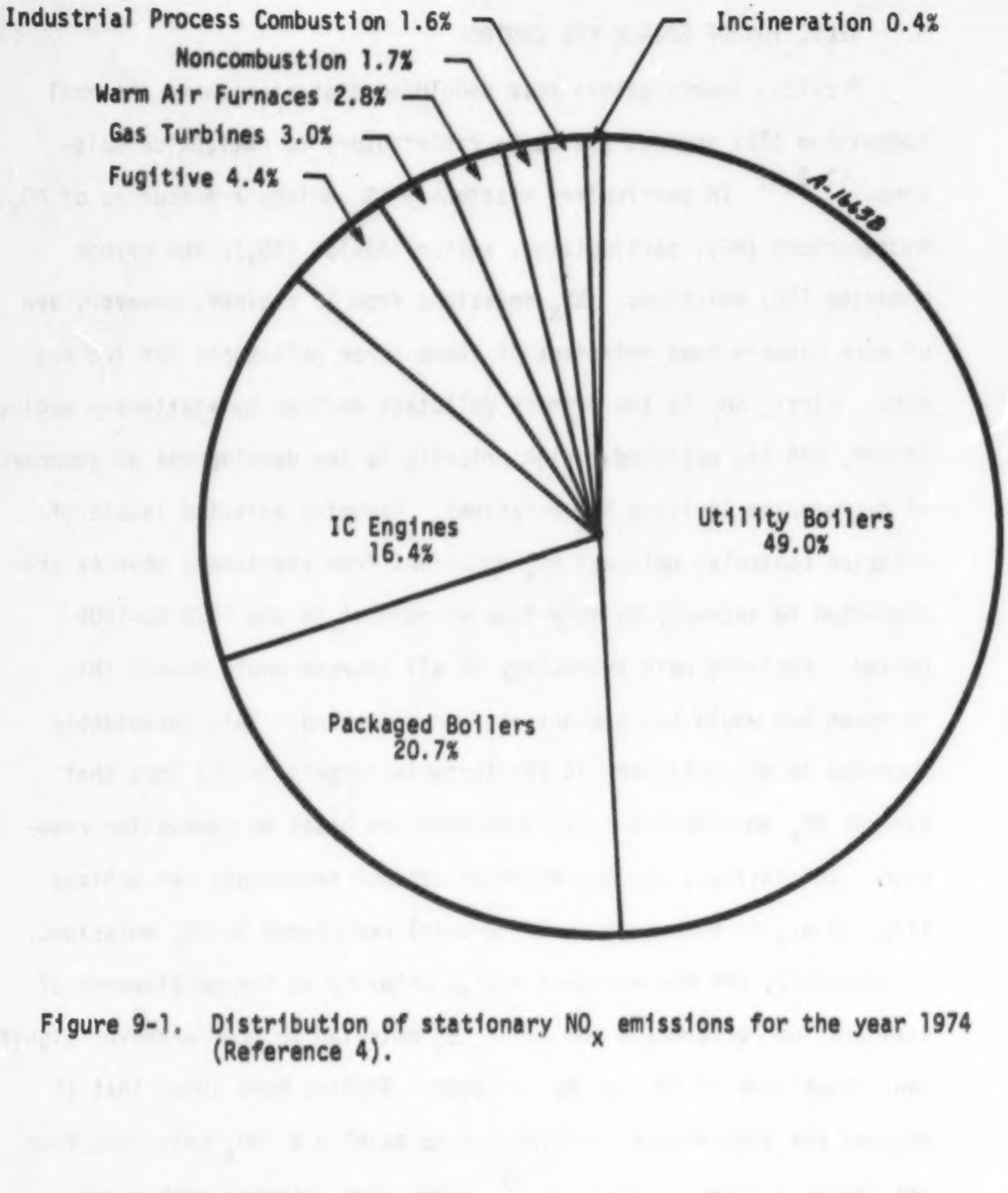

Figure 9-1. Distribution of stationary $NO_x$ emissions for the year 1974 (Reference 4).

exceeded only by utility and packaged boilers.

An inventory of emissions from installed stationary engines was computed based on the information presented in summary form in Table 9-1.(5) As a group, stationary IC engines (based on 1975 data) currently account for 3 to 9 percent of the $NO_x$, carbon monoxide (CO), and hydrocarbons (HC) emitted from all sources, and 9 to 14 percent of those emitted from stationary sources. This table also shows the percentage contribution to nationwide totals from installed engines as a function of their size and the type of fuel they consume. Table 9-2(6) shows the emission factors used to generate Table 9-1. Annual production rates are estimated in Table 9-1 to indicate the potential number of sources that could be affected by New Source Performance Standards (NSPS).

Table 9-3(7) presents a clearer picture of the relationship between the number of potentially controllable sources and their contributions to the nationwide inventory from currently installed units.* This table also shows that numerous small engines (nearly 13 million units of 1- to 100-hp) are the most significant contributors of HC and CO emissions from IC engines. (Note that nearly 80 percent of the HC emissions from engines smaller than 350 CID/cyl are methane, a noncriteria pollutant.) Therefore, it can be concluded that $NO_x$ emissions constitute the most significant pollutant emitted by stationary engines since three-quarters of these emissions are emitted by large-bore (greater than 350 CID/cyl) engines.

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*Table 9-3 includes a separate row of emission estimates for engines larger than 350-cubic-inch displacement per cylinder (CID/cyl). As is shown in Section 3, it is more meaningful to discuss the applications and emissions of large IC engines on the basis of displacement per cylinder rather than on horsepower.

TABLE 9-1. NATIONWIDE EMISSIONS FROM INSTALLED IC ENGINES
(Percent of Total Emitted in U.S. Each Year)[a]

| Fuel | hp Range | Annual Production, Units[b] | $NO_x$ | CO | $HC_T$ |
|---|---|---|---|---|---|
| Diesel | 20 - 100 | 39,000 | 0.36 | 0.029 | 0.062 |
| | 101 - 500 | 14,000 | 0.55 | 0.45 | 0.095 |
| | >500 | 3,400 | 0.48 | 0.016 | 0.033 |
| | Subtotal | 56,400 | 1.39 | 0.09 | 0.19 |
| Natural Gas | <500 | 5,400 | 1.93 | 0.107 | 0.81 |
| | >500 | 600 | 4.16 | 0.229 | 1.73 |
| | Subtotal | 6,000 | 6.10 | 0.336 | 2.54 |
| Dual-Fuel | Included in Diesel | | 0.28 | 0.02 | 0.11 |
| Gasoline | <15 | 12,600,000[c] | 0.16 | 1.84 | 0.56 |
| | 15 - 99 | 85,000 | 0.31 | 0.81 | 0.29 |
| | >100 | 10,000 | 0.11 | 0.31 | 0.10 |
| | Subtotal | 12,600,000<br>+ 95,000 | 0.58 | 2.96 | 0.95 |
| | Total | 12,600,000<br>+ 157,400 | | | |
| Percent All Sources | | | 8.4 | 3.4 | 3.8 |
| Percent Stationary Sources | | | 13.7 | 11.0 | 8.8 |
| In Mass Units ($10^6$ metric tons/yr) | | | 2.0 | 3.6 | 0.9 |

[a] Total U.S. emission from EPA Nationwide Air Pollutant Inventory for 1975(5)

[b] Based on estimates of average hp of engines used in each application

[c] Includes all engines in this size category (mobile and stationary). Listed separately in the totals because of the unique nature of this group

TABLE 9-2. EMISSIONS FACTORS FOR INVENTORY ON TABLE 1,[a] g/hp-hr (Reference 6)

| Fuel | | $NO_x$ | CO | $HC_T$ |
|---|---|---|---|---|
| Gasoline | >15 hp | 8.85 | 102 | 8.38 |
| | <15 hp | 5.63 | 295 | 20.5 |
| Diesel | >500 hp[b] | 12.9 | 1.8 | 0.43 |
| | <500 hp[c] | 12.4 | 4.47 | 2.12 |
| Natural gas | | 11.5 | 2.81 | 4.86 |
| Dual-Fuel | | 8.2 | 2.0 | 3.1 |

[a] Emission factors for gasoline and diesel engines are modal averages; those for natural gas and dual-fuel are for rated conditions.

[b] Based on an average of rated condition levels from engines considered

[c] Weighted average of two- and four-stroke engines. Weighting factors = 2/3 for four-stroke and 1/3 for two-stroke

## TABLE 9-3. EMISSIONS FROM IC ENGINES BY SIZE AND ANNUAL PRODUCTION[a]

| Annual Production | | | Emissions from Installed Engines[b] (% U.S. Total) | | | | | |
|---|---|---|---|---|---|---|---|---|
| Size | Units/yr | Increment | $NO_x$ | Increment | CO | Increment | $HC_T$ | Increment |
| >350 CID/cyl[c] | 1,600 | | 4.83 | | 0.26 | | 1.86 | |
| >500 hp | 4,000 | 2,400 | 4.92 | 0.09 | 0.26 | 0.003 | 1.87 | 0.01 |
| >100 hp | 28,000 | 24,000 | 5.59 | 0.66 | 0.59 | 0.33 | 2.07 | 0.20 |
| >15 hp | 152,000 | 124,000 | 6.26 | 0.67 | 1.43 | 0.83 | 2.42 | 0.35 |
| All | 12,752,000 | 12,600,000 | 6.42 | 0.16 | 3.27 | 1.84 | 2.98 | 0.56 |
| Total | ($10^6$ metric tons/year) | | 1.53 | | 3.47 | | 0.72 | |
| All Sources | ($10^6$ metric tons/year)[d] | | 23.8 | | 105.8 | | 24.2 | |
| Stationary Sources | ($10^6$ metric tons/year)[d] | | 14.5 | | 32.6 | | 10.4 | |

[a] Compiled from Table 9-1

[b] Excludes emissions from natural gas engines under 500 hp (insignificant future impact projected from new engines due to declining sales)

[c] Cubic Inch Displacement per cylinder. All gas and dual-fuel engines >500 hp are taken to be greater than 350 CID/cyl. Of the 3400 diesel units >500 hp, 1000 are assumed to be greater than 350 CID/cyl. These large bore diesels contribute 80 percent of the emissions from diesel units >500 hp.

[d] From EPA Nationwide Air Pollutant Inventory for 1975(7).

Other studies have investigated the emissions of various stationary sources to aid in establishing a priority for setting standards of performance. For example, the Research Corporation of New England determined the effect that standards of performance would have on nationwide emissions of particulates, $NO_x$, $SO_x$, HC, and CO from stationary sources.(8) As per EPA-450/3-78-019, "Priorities for New Source Performance Standards under the Clean Air Act Amendments of 1977," sources were ranked according to the impact, in tons per year, that standards promulgated in 1980 would have on emissions in 1990. This ranking placed spark ignition IC engines second and compression ignition IC engines third on a list of 32 stationary $NO_x$ emission sources. Consequently, stationary IC engines have been selected for development of standards of performance.

In a subsequent study, Argonne National Laboratory used the results of the TRC study to develop a priority listing for setting NSPS.(9) In developing this list, source screening factors were used to aid in establishing these control priorities. Factors considered were:

- Type, cost, and availability of control technology
- Emission measurement methods and applicability
- Enforceability of regulations
- Source location and typical source size
- Energy impact
- Impact on scarce resources
- Other environmental media constraints

The study found that even with the application of maximum NSPS control efforts, a significant increase of more than 40 percent in $NO_x$ emissions occurs in the 1975-to-1990 period. Furthermore, the study concluded that the control of internal combustion engines emissions is a matter of high priority.

Other factors favoring the control of IC engines are summarized briefly below:

- Control techniques for $NO_x$ emissions have been shown to be effective and applicable to installed IC engines. These techniques can reduce $NO_x$ emissions from 40 to 60 percent on the average (see Section 4.0).
- No federal, state or local $NO_x$ standards exist (with the exception of Los Angeles and Chicago). Therefore, since engines are manufactured for a variety of dispersed applications, a single national standard is preferable.
- IC engines compete with gas turbines in certain applications. Since NSPS are currently being developed for gas turbines, the absence of standards for IC engines may result in a shift away from gas turbines to IC engines. This could cause greater $NO_x$ emissions from both sources than if no standard were applied to gas turbines, since IC engines emit $NO_x$ at greater rates than gas turbines.

Furthermore, as shown in Section 3.0, sales of large-bore engines, primarily for oil and gas exploration, have been substantial during the past five years, and are anticipated to continue and possibly increase. Stationary IC engines, therefore, are significant contributors to total nationwide emissions of $NO_x$. Consequently, based on all these factors, stationary IC engines have been selected for development of standards of performance.

## 9.2 SELECTION OF POLLUTANTS

### Oxides of Nitrogen

Stationary engines emit the following pollutants: $NO_x$, CO, HC, particulates, and $SO_x$. As Table 9-3 indicates, the primary pollutant emitted by stationary engines is $NO_x$, accounting for over six percent (or 16 percent of all stationary sources) of the total U.S. inventory of $NO_x$ emissions. This table also illustrates that large-bore engines emitted three-fourths of these $NO_x$ emissions. It will be shown in Section 4.0 that the control technology exists to effectively reduce $NO_x$ emissions from large-bore engines. Furthermore, $NO_x$ emissions are projected to increase despite promulgation of all possible New Source Performance Standards. Therefore, $NO_x$ emissions from stationary engines have been selected for control by means of NSPS.

### Hydrocarbons and Carbon Monoxide

Table 9-3 also shows that stationary IC engines emit substantial quantities of CO and HC as well. Numerous small (1-100 hp) spark ignition engines, which are similar to automotive engines, account for about 20 percent of the uncontrolled HC emissions and about 80 percent of the uncontrolled CO emissions. However, as mentioned in Section 9.3, the large annual production of these small spark ignition engines (approximately 12.7 million) makes enforcement of a new source performance standard for this group difficult.

An additional factor in considering CO and HC control is that inherent engine characteristics result in a trade-off between $NO_x$ control and control of CO and HC. A detailed discussion of the trade-off can be found in Section 9.4. In some cases, particularly naturally aspirated gas engines, the application of $NO_x$ emission control techniques could cause increases in CO and HC emissions. This increase in CO and HC emissions is strictly a function of the engine operating position relative to stoichiometric conditions, not the $NO_x$ control technique. Any increase in CO and HC emissions, however, represents an increase in unburned fuel and hence a loss in efficiency. Since IC engine manufacturers compete with one another on the basis of engine operating costs, which is primarily a function of engine operating efficiency, the marketplace will effectively ensure that CO and HC emissions are as low as possible following the application of $NO_x$ control techniques. In addition, promulgation of CO and HC emissions standards of performance could, in effect, preclude significant $NO_x$ control. CO emissions, which are primarily a function of oxygen availability and only secondarily of temperature, show a pronounced rise as the mixture becomes richer than stoichiometric, but little variation as it becomes leaner. Carbureted engines, however, which are beset by large variations in cylinder-to-cylinder air-to-fuel ratios, must operate near the stoichiometric ratio to ensure that no individual cylinder receives a charge which is too lean to ignite (i.e., exceeds the lean misfire limit). Consequently, increasing the air-to-fuel ratio to near stoichiometric to reduce the CO emissions increase has the effect of also limiting the $NO_x$ emissions reduction.

These and other factors discussed in Section 9.3 led to the recommendation of a $NO_x$ NSPS for large-bore engines but not for HC and CO emissions since:

- The IC engines which emit significant quantities of $NO_x$ are, with some exceptions, low emitters of HC and CO
- Many of the $NO_x$ reduction techniques discussed in Section 4.4 cause little or no increase in the already low HC and CO emissions rates from most large-bore engines
- Individual engines can cause violation of the National Ambient Air Quality Standards for HC only under worst-case atmospheric conditions, and then only very close to the source (less than 0.3 km)
- No controls for HC used in conjunction with $NO_x$ controls have been demonstrated which reduce the already low nonmethane HC emissions from large-bore engines

Particulate

No standards of performance are recommended for either particulate emissions or visible emissions (plume opacity). This recommendation stems from the following considerations:

- Virtually no data are available on particulate emission rates from stationary engines because it is so difficult, expensive, and time-consuming to measure particulates, especially when done in strict compliance with EPA Method 5 sampling techniques
- It would be very expensive to enforce a standard on required measurements for particulates in compliance testing which would be in accordance with EPA Method 5
- It is believed that particulate emission from stationary engines are relatively unimportant because the plumes from most of these engines are not now visible

## $SO_x$

Sulfur oxides ($SO_x$) emissions are strictly dependent upon the sulfur contained in the fuel. Thus, annual sulfur oxide emissions from an engine depend on the percent sulfur in the fuel and the fuel consumption of the engine during that year. Most engines burn low-sulfur fuels and will continue to do so since crude and residual fuels must be treated to remove the salts from the fuels, and inhibitors must be added to prevent the vanadium in the fuel from corroding IC engine components. Treatment facilities exist, but their function is minimal. The primary reason for the shift away from the treatment of crude and residual fuels is one of economics. In today's market, it simply costs more to buy and treat the crude and residual oils than to purchase and burn the distillate oils.

The cost of flue gas desulfurization for IC engines does not appear to be reasonable from an economic viewpoint. Therefore, the only viable means of controlling $SO_x$ emissions would be combustion of low-sulfur fuels. If users in urban or $SO_x$-sensitive areas decide to buy new engines and to use crude or residual oil as a fuel with these engines, then the local air pollution authorities could impose fuel restrictions on these engines. Such fuel restrictions would be entirely independent of the standards of performance from both a technological and enforcement viewpoint. That is, the absence of federal emission limits on $SO_x$ would not prevent a local air pollution control district from setting such a standard since the engine would not have to be changed in order to meet the local standard. Thus, standards of performance are not recommended for $SO_x$ emissions.

## 9.3 SELECTION OF AFFECTED FACILITIES

In sections 1.0 and 2.0 it was shown that $NO_x$ emissions constitute the most significant pollutant emitted by stationary IC engines, and that large-bore (greater than 350 CID/cyl) engines account for over 75 percent of all $NO_x$ emission from stationary engines. This section will establish criteria that define which large diesel, dual-fuel, and natural gas engines (referred to as "affected facilities") are to be affected by the proposed standards of performance. The objective here is to apply standards of performance to significant sources of $NO_x$ emissions.

Thus, the following sections will present and explain the criteria that define affected facilities after considering the applications served by stationary engines, the number of units produced annually, and the incremental $NO_x$ contributed by the annual production. The following discussions are subdivided by the three operational fuel types: diesel, dual-fuel, and natural gas. As will be discussed in the following paragraphs, this classification separates large-bore engines into three relatively distinct categories of engine applications. Initially, large-bore engines will be defined as those exceeding 350 CID/cyl. Then, if necessary, other criteria will be presented and explained to define affected diesel, dual-fuel, and natural gas engines.

The following discussion summarized an extensive study of the applications of large-bore engines. Many of the conclusions presented here are based on information concerning engine sales and applications during the past five years. This information was voluntarily submitted by engine manufacturers in response to Section 114 requests for information.(11) This information has not been cited for particular manufacturers since it is considered proprietary by the manufacturers.

## Affected Diesel Engines

The primary high usage (large emissions impact), domestic application of large-bore (i.e., greater than 350 CID/cyl) diesel engines during the past five years has been for oil and gas exploration and production. These and other applications are illustrated in Figure 9-2.(12-17) As this figure shows, the market for prime (continuous) electric generation and other industrial applications all but disappeared after the 1973 oil embargo, but was quickly replaced by sales of standby electric units for building services, utilities, and nuclear power stations. The rapid growth in the oil and gas production market occurred because diesel units are being used on oil drilling rigs of various sizes. Sales of engines to export applications have also grown steadily since 1972, and are not a major segment of the entire sales market.

Medium-bore (from 35 to 350 CID/cyl) as well as large-bore engines are sold to oil and gas exploration, standby service, and other industrial applications. Furthermore, manufacturers of medium- and large-bore engines often compete for the same applications, although, in general, medium-bore engines have a cost advantage (lower $/hp). This is because the higher initial costs for a large-bore, heavy-duty, continuous-service engine more than offset their lower maintenance and fuel costs. This overlap in sizes is best illustratred in Figure 9-3 which shows a considerable number of medium- and large-bore engines in the 500- to 2000-horsepower range. Figure 9-4 shows the displacement per cylinder that

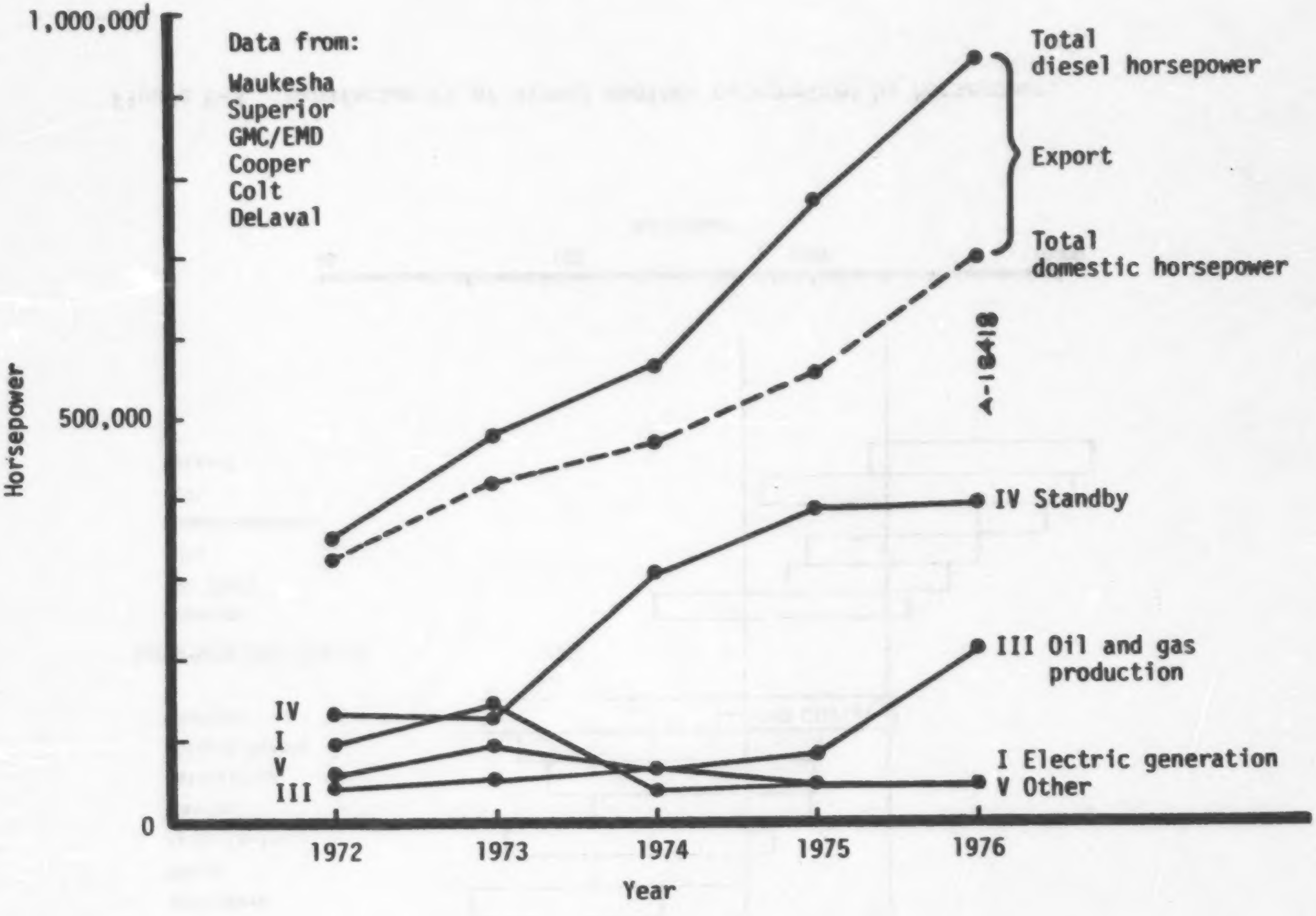

Figure 9-2. Sales of large-bore (>350 CID/cyl) diesel horsepower (from References 12-17).

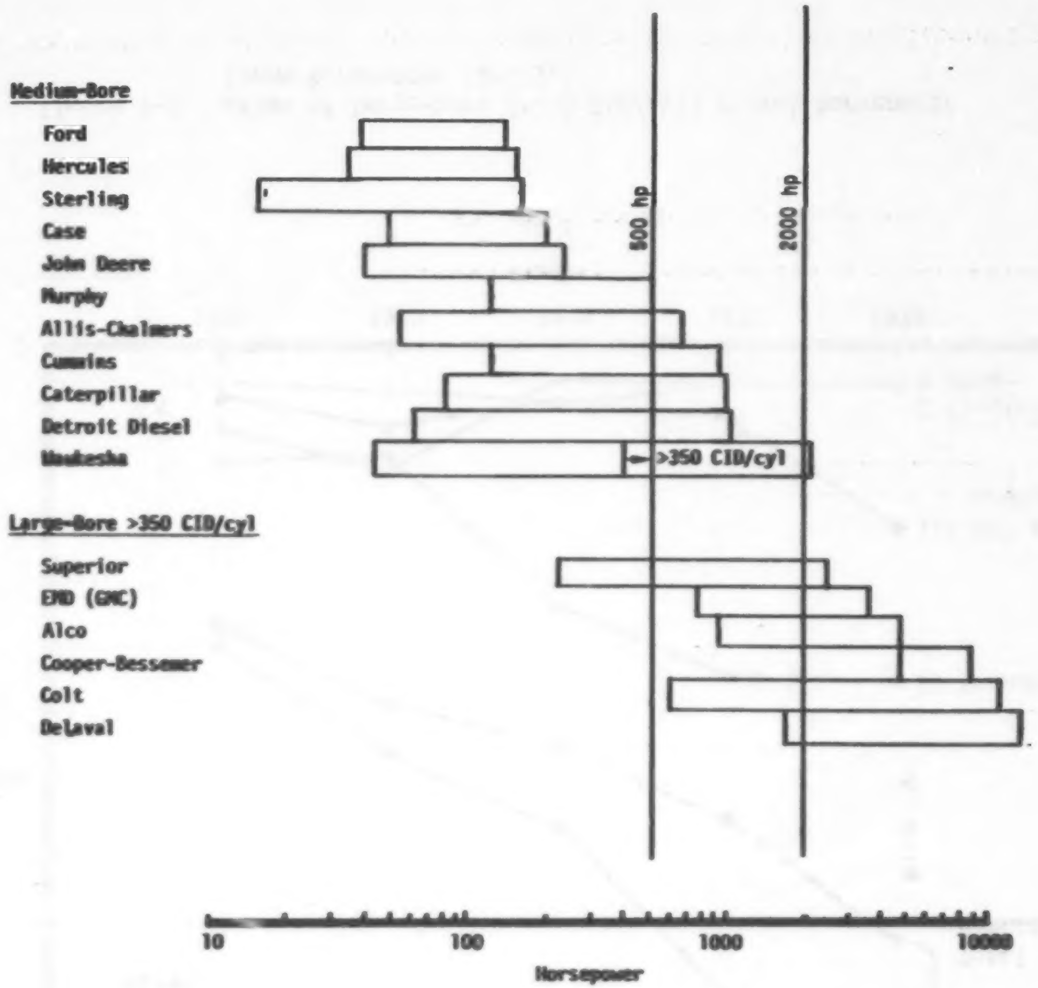

Figure 9-3. Manufacturers of diesel engines categorized by horsepower.

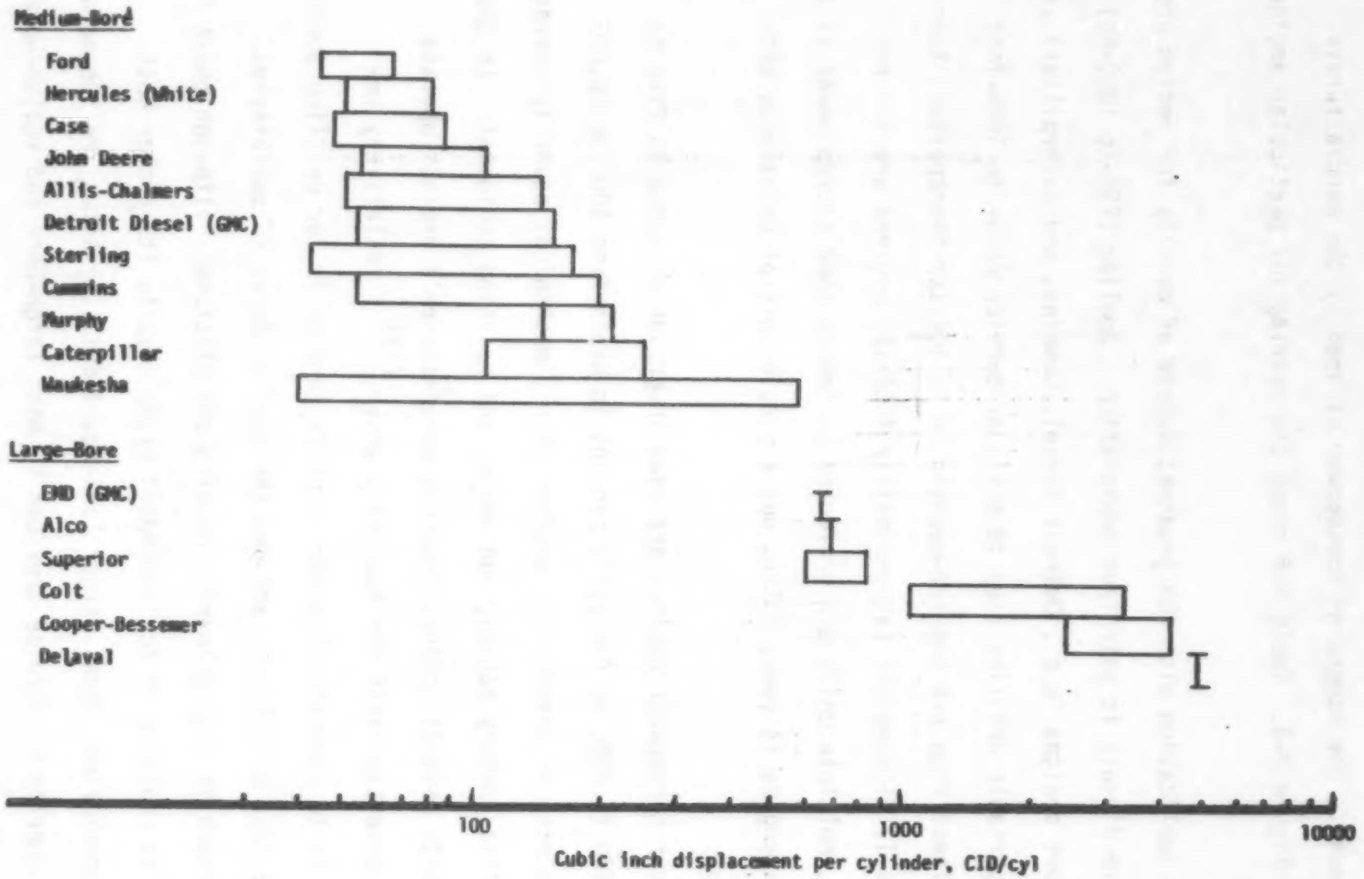

Figure 9-4. Manufacturers of diesel engines categorized by cubic-inch displacement per cylinder.

corresponds to the ranges of horsepower offered by the manufacturers shown in Figure 9-3. Table 9-4 shows the overlap for particular engine models.

The application with the greatest degree of overlap for medium and large-bore diesels is petroleum exploration. Smaller (250-to 1000-hp) medium-bore designs (e.g., Detroit Diesel, Cummins, and Caterpillar) are used on portable drilling rigs to drill or service 2500- to 5000-foot wells. These rigs are trailer-mounted or helicopter-transported; therefore, small, lightweight (approximately 4000-lb) engines are favored. In addition, multiple units are preferred to insure some backup power in the event one engine is down, ruling out a single unit of comparable total horsepower.

Larger horsepower engines are used in groups of three to five to provide 800 to 3000 hp for wells ranging in depth from 5000 to 25,000 feet. On most of these rigs, engines supply mechanical power to operate the drilling (rotary table), mud pumps, and hoisting equipment. In the larger units several engines from one manufacturer's engines operate pumps or generator sets for auxiliary power.(18) A relatively new approach is to generate AC power, rectify some of it for drilling power (variable load DC motors), and use the rest to drive AC auxiliaries. This approach is used primarily on offshore platforms, although there is interest in applying it to land-based sites despite its higher cost.

In conclusion, then, larger land-based drilling sites are the major areas of overlap of service provided by both large-bore and medium-bore manufacturers. These applications and baseload electric generation (to a lesser extent, since horsepower sales are small) have the most signifi-

## TABLE 9-4. COMPARISON OF WAUKESHA, WHITE SUPERIOR, AND MEDIUM-BORE ENGINE MODELS GREATER THAN 500 HORSEPOWER

| Manufacterer | Model | Number of Cylinders | Displacement per Cylinder, CID/CYL | Displacement, cubic inches | Type | Continuous Rated Horsepower | rpm |
|---|---|---|---|---|---|---|---|
| Waukesha | VHP | 6 | 482 | 2,896 | NA | 411 | 1,200 |
| Waukesha | VHP | 6 | 482 | 2,896 | TC | 561 | 1,200 |
| Waukesha | VHP | 6 | 482 | 2,896 | TC,AC | 702 | 1,200 |
| Waukesha | VHP | 6 | 556 | 3,335 | TC,AC | 808 | 1,200 |
| Waukesha | VHP | 12 | 482 | 5,792 | NA | 818 | 1,200 |
| Waukesha | VHP | 12 | 482 | 5,792 | TC | 1,123 | 1,200 |
| Waukesha | VHP | 12 | 482 | 5,792 | TC,AC | 1,403 | 1,200 |
| Waukesha | VHP | 12 | 556 | 6,670 | TC,AC | 1,616 | 1,200 |
| Waukesha | VHP | 16 | 556 | 8,894 | TC,AC | 2,154 | 1,200 |
| Cummins | VTA-1710-P<br>KTA | 12<br>12 | 143<br>192 | 1,710<br>2,300 | TC,AC<br>TC,AC | 547<br>900 | 1,800<br>1,900 |
| Caterpillar | D398TA<br>D399TA | 12<br>16 | 245<br>245 | 2,945<br>3,927 | TC,AC<br>TC,AC | 750<br>1,000 | 1,200<br>1,200 |
| Detroit Diesel | 12V-149<br>16V-149 | 12<br>16 | 149<br>149 | 1,788<br>2,384 | TC,AC<br>TC,AC | 810<br>1,080 | 1,800<br>1,800 |
| Superior Division/Cooper | 40-X-6<br>PTD5-6 | 6 | 687/596 | 4,120/3,575 | TC,AC | 790/675 | 1,000 |
|  | 40-X-8<br>PTDS-8 | 8 | 687/596 | 5,493/4,767 | TC,AC | 820/945 | 900 |

cant $NO_x$ emissions impact because they are high usage (approximately 6000 hr/yr). However, a greater-than-350-CID/cyl definition of affected facilities would result in some manufacturers (e.g., Waukesha) being subject to control technology development costs, while medium-bore engines (of same power, but more cylinders) serving identical applications would not incur these costs. This is clearly undesirable since this definition would unfairly place some large-bore engines in a less competitive position than similarly sized (by horsepower), smaller-bore designs.

On the other hand, applying the standards of performance to medium-bore engines serving the same applications as large-bore designs would increase the number of affected facilities from about 200 to about 2000 units per year (based on 1976 sales information) but consequently further reduce $NO_x$ emissions. Medium-bore sales accounted for significant $NO_x$ emissions in 1976 (approximately 12,500 megagrams). It is estimated that approximately 25 percent, or about 500 of these units, in high usage applications accounted for most of the medium-bore $NO_x$ emissions, since most of the remainder of these units were sold as standby generator sets. Though the potential achievable $NO_x$ reduction is significant, considering this large number, and the remoteness and mobility of petroleum applications, this alternative would create serious enforcement difficulties. Additionally, this alternative causes the standard to apply to lower power engine models with fewer number of cylinders competing in different stationary markets with other unregulated engines. Consequently, a definition is required that distinguishes between large-bore engines that compete with medium-bore high power engines used for baseload electrical generation from large-bore engines that compete solely with other large-bore engines.

One approach would be to define diesel engines covered by standards of performance as those exceeding 560 CID/cyl. This alternative would exclude engines presently manufactured by Waukesha as well as those produced by Caterpillar, Detroit Diesel, and Cummins. This definition, however, shifts the area of overlap in horsepower between regulated and unregulated engines to other large-bore diesel manufacturers. This situation is depicted in Figure 9-5, which illustrates the relationship between displacement per cylinder and rated (continuous) horsepower. All Waukesha engines are excluded above the 560 CID/cyl limit. However, Superior's diesel engines ranging in size from 596- to 825-CID/cyl would be subject to standards. These engines compete in very few cases with Waukesha diesel engines. Raising the limit to 700-CID/cyl would exclude Superior engines in the 500- to 100-hp range, but it would also exclude EMD and Alco models, which compete with Colt (700-CID/cyl, hence regulated) in the 1000- to 3000-hp range. Establishing a 560-CID/cyl definition, therefore, appears to be a viable method of excluding engines which compete with medium-bore designs without introducing a significant overlap problem at a different power level.

After considering the sizes and displacements offered by each diesel manufacturer and the applications served by diesel engines, a 560-CID/cyl definition was selected as a reasonable approach for separating large-bore engines that compete with medium-bore engines from large-bore engines that compete solely with each other. This cylinder displacement size was chosen because engines below this size are generally used for

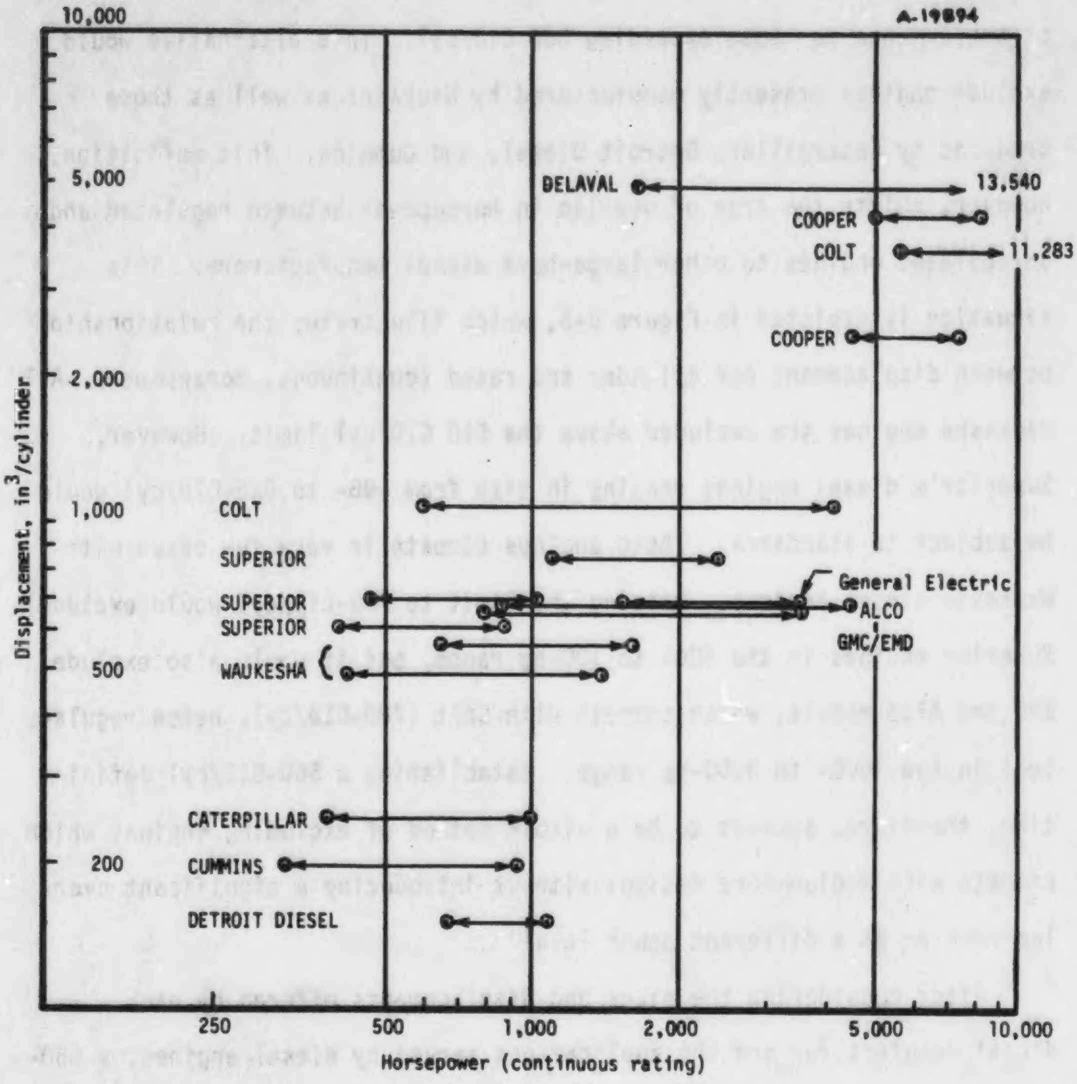

**Figure 9-5. Displacement per cylinder versus continuous rated horsepower for diesel engines.**

different applications than those above it. Therefore, it is recommended that diesel engines greater than 560 CID/cyl be affected by standards of performance.

## Affected Dual-Fuel Engines

The concept of dual-fuel operation was developed to take advantage of both compression ignition performance and inexpensive natural gas. These engines have been used almost exclusively for prime electric power generation. Figure 9-6(19-22) illustrates, however, that shortages of natural gas and the 1973 oil embargo have combined to significantly reduce the sales of these engines in recent years. The few large-bore units that were sold (11 in 1976) were all greater than 350 CID/cyl. In fact, with the exception of Superior Division/Cooper and Stewart-Stevenson (modified Detroit Diesel engine) products, all were greater than 500 horsepower and 1000 CID/cyl as shown in Figures 9-7 and 9-8. Moreover, nearly all of the dual-fuel engines sold since 1972 have been larger than 1000 hp. Only Stewart-Stevenson manufactures dual-fuel engines less than 560 CID/cyl. Sales of these units are less than 100 units per year and about 70 percent of these are exported.(23)

Although a greater-than-350-CID/cyl limit would subject nearly all new dual-fuel sources to standards of performance (only engines manufactured by Stewart-Stevenson would be excluded), it is recommended that the definition chosen to define affected diesel engines (560 CID/cyl) also be applied to dual-fuel engines. The reason is that supplies of natural gas are likely to become even more scarce, possibly causing recently installed or future dual-fuel units to convert to diesel fuel operation. Any additional diesel engines that would be created by con-

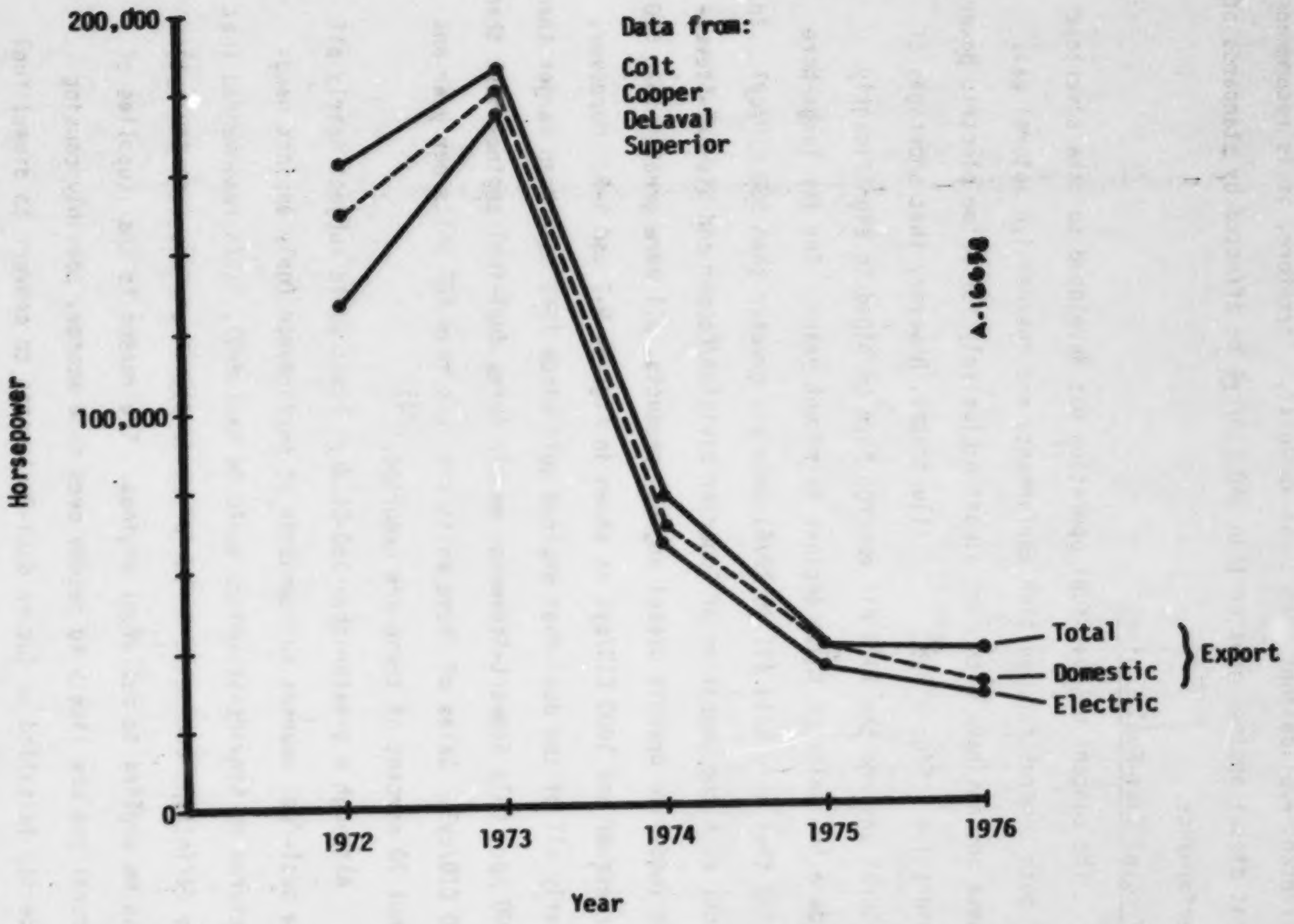

Figure 9-6. Sales of dual-fuel horsepower during the past 5 years (from References 19 to 22).

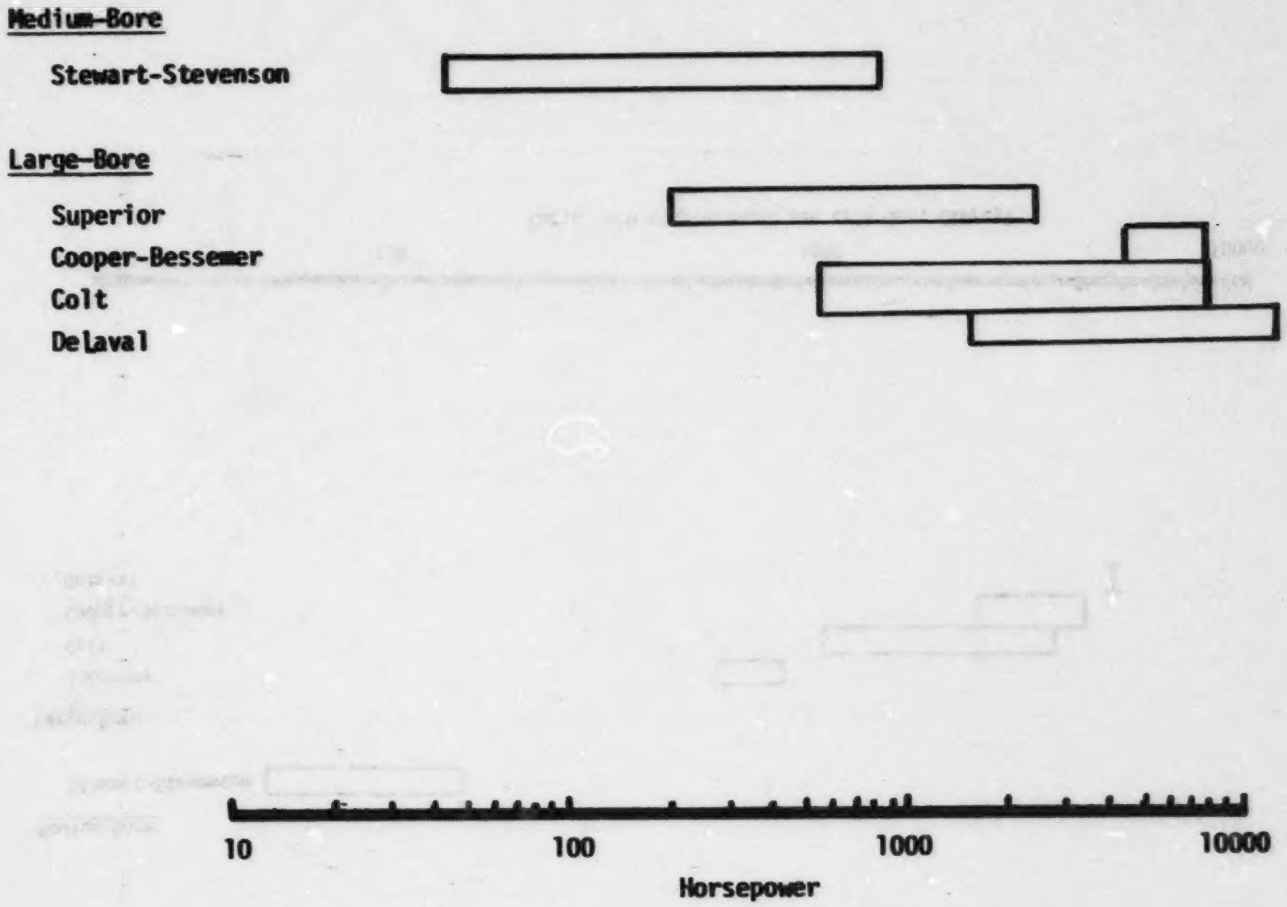

Figure 9-7. Manufacturers of dual-fuel engines categorized by horsepower.

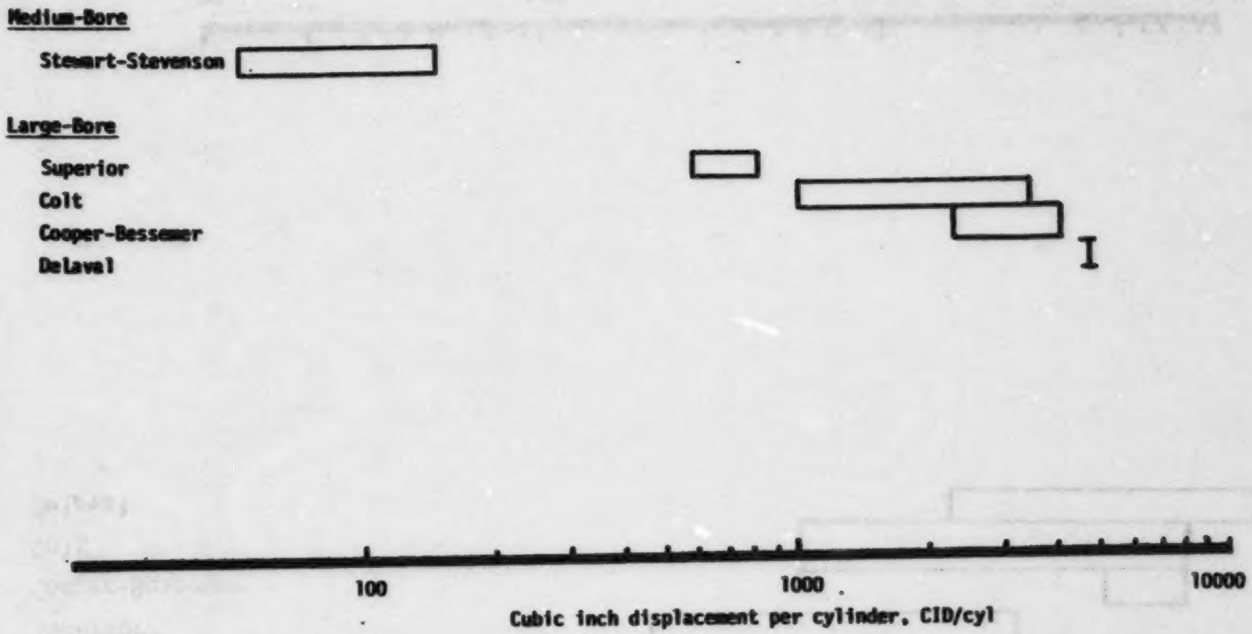

Figure 9-8. Manufacturers of dual-fuel engines categorized by cubic-inch displacement per cylinder.

version from dual-fuel operation should be subject to the same regulations applicable to other large diesel engines.

Affected Gas Engines

The primary application of large (greater than 350 CID/cyl) gas engines during the past five years has been for oil and gas production. The primary uses are to power gas compressors for recovery, gathering, and distribution. Figure 9-9(24-29), based on manufacturer's data from response to the June 16, 1976 Section 114 requests for information, illustrates that 75 to 80 percent of all gas engine horsepower sold during the past five years was used for these applications.

During this time sales to pipeline transmission applications declined. Pipeline applications combined with standby power, electric generation, and other services (industrial and sewage pumping). These other applications accounted for the remaining 20 to 25 percent of horsepower sales. The growth of oil and gas production applications during this period corresponds to the increasing efforts to find new, or recover marginal, gas reserves and distribute them to the existing pipeline transmission network.

Figure 9-10 illustrates the number of gas engines sold for five size groups during the past five years. The large number of smaller-than-500-hp engines that were sold during this period are one or two cylinder engines used on oil well beam pumps and for natural gas well recovery and gathering. Most of the other larger gas engines that were sold during this period ranged from 500 to 2000 hp. In 1976, approximately 400 engines in this size range were sold, primarily for oil and gas production (see Figure 9-9). Most of these gas engines were manufactured by Cater-

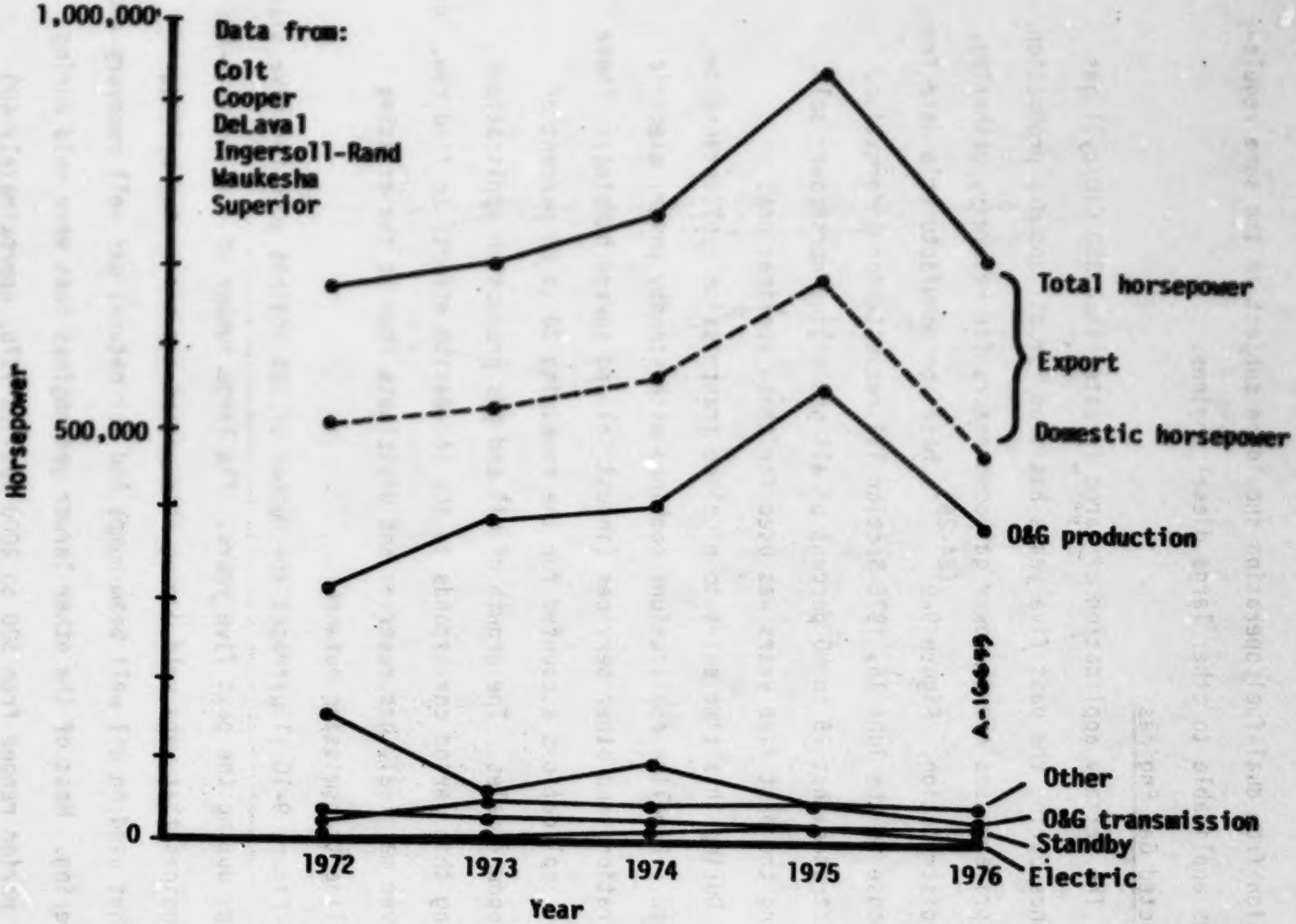

Figure 9-9. Sales of gas engine horsepower by application for the past 5 years (absolute levels shown for domestic applications from References 24 to 29).

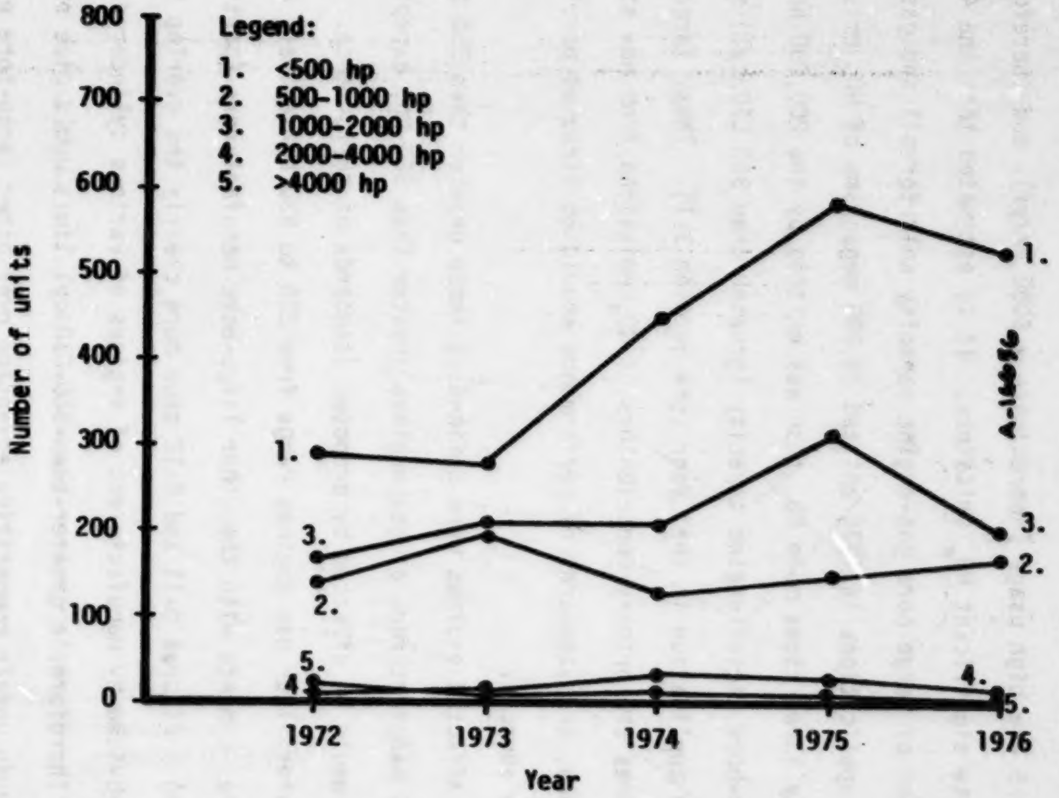

Figure 9-10. Size distribution of gas engines sold during the past 5 years.

pillar, Cooper, Waukesha, and White Superior.

With the exception of standby service, all the applications of Figure 9-9 are high usage (approximately 6000 hr/yr), and therefore, contribute significant $NO_x$ emissions. It is estimated that the 400,000 horsepower of large-bore gas-engine capacity sold for oil and gas production applications in 1976 emitted 34,900 megagrams of $NO_x$ emissions, or nearly three times more $NO_x$ than was emitted by the 200,000 horsepower of large-bore diesel engine capacity (greater than 350 CID/cyl) sold for the same application in that year (see Section 3.1). Thus, large-bore gas engines are primary contributors of $NO_x$ emissions from new stationary IC engines, and standards of performance should be directed particularly at these sources.

If affected engines were defined as those greater than 350 CID/cyl, then all manufacturers of gas engines greater than 500 hp, except Caterpillar, would be affected by proposed standards of performance. However, large Caterpillar gas engines range from 225 to 930 horsepower, and therefore, compete with the other large-bore manufacturers (particularly Waukesha). Figures 9-11 and 9-12 show more clearly the overlap in horsepower provided by manufacturers of engines of various cylinder displacements. Therefore, a greater-than-350-CID/cyl limit would give one manufacturer an unfair competitive advantage over other large-bore engine manufacturers. Thus, although a greater-than-350-CID/cyl limit would subject most significant gas engine sources of $NO_x$ emissions to potential standards of performance, this definition should be revised based on the following considerations:

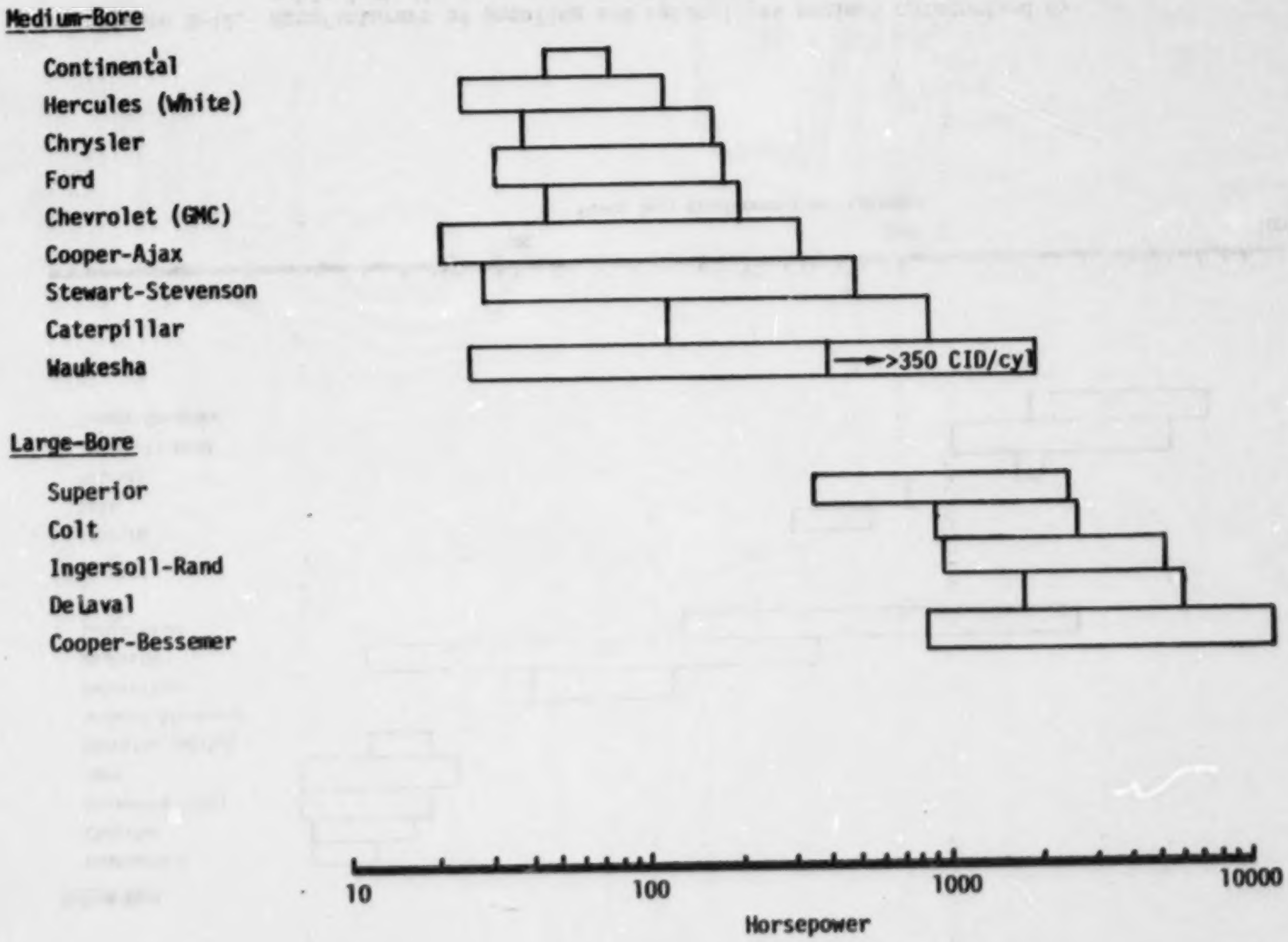

Figure 9-11. Manufacturers of gasoline and natural gas engines categorized by horsepower.

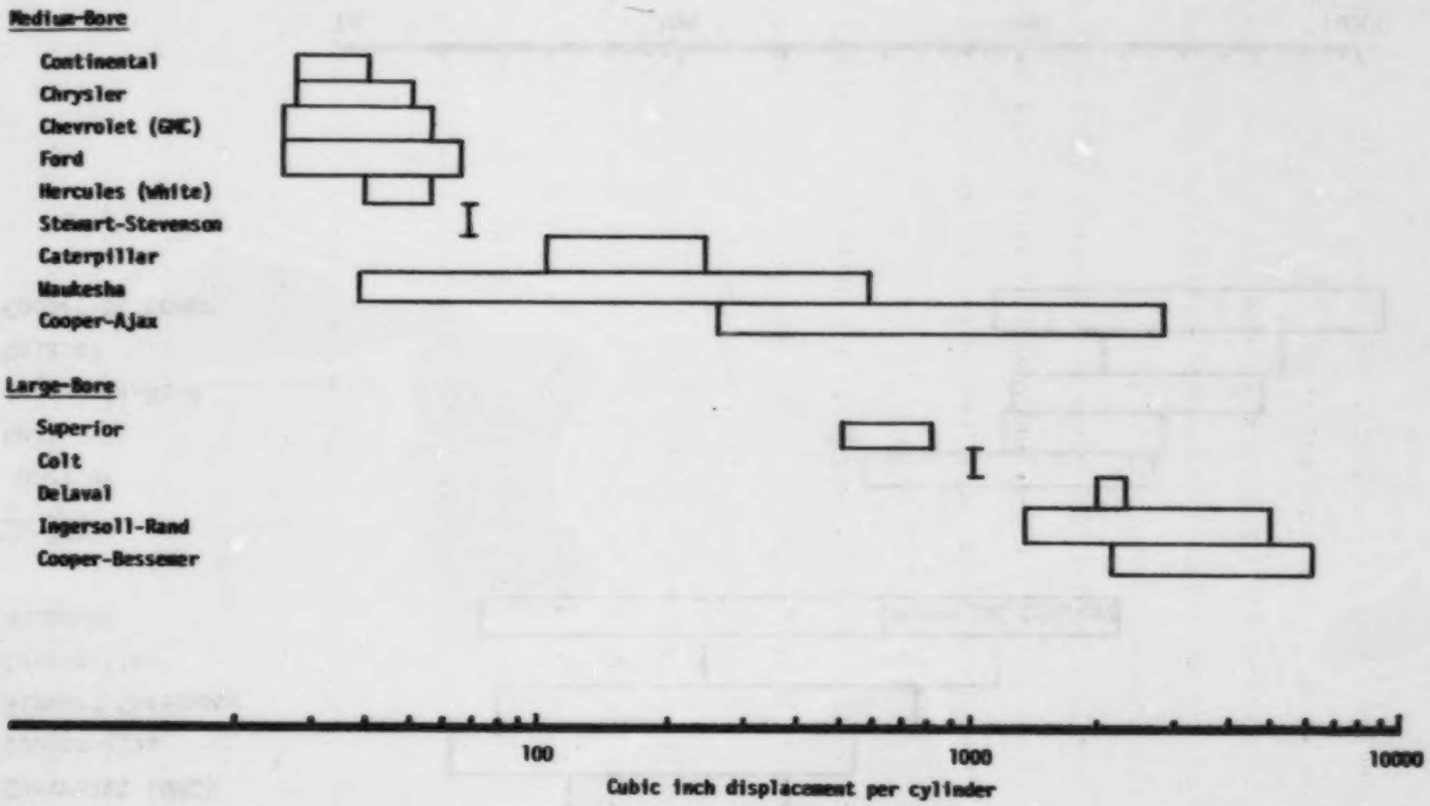

Figure 9-12. Manufacturers of gasoline and natural gas engines categorized by cubic-inch displacement per cylinder.

- The greater-than-350-CID/cyl definition excludes the only other manufacturer (Caterpillar) of gas engines greater than 500 hp. Caterpillar gas engines compete directly with the large gas engines manufactured by Cooper, Waukesha, and White Superior, which would be regulated.
- No emissions have been measured or control techniques demonstrated for 1- and 2-cylinder engines which would be included in potential standards of performance by the existing greater than 350 CID/cyl limit.

The first observation suggests that the definition should be lowered, or another definition adopted, to include the large Caterpillar engines that compete in identical applications with Cooper, Waukesha, and White Superior units. Although Caterpillar has not reported controlled emissions data for their gas engines, control techniques have been demonstrated on other similar gas engines and should be effective when applied to Caterpillar engines, since they are all similar in design (i.e., carbureted and gas injected engines that are either turbocharged and aftercooled or naturally aspirated).

Table 9-5 compares large Caterpillar gas engines with Waukesha models that are greater than 350 CID/cyl. As this comparison illustrates, Caterpillar engines with smaller displacements per cylinder and greater numbers of cylinders serve about the same power range as do the larger Waukesha engines. On the basis of this table, either of the following two steps would subject Caterpillar gas engines to potential standards of performance:

## TABLE 9-5. COMPARISON OF LARGE CATERPILLAR GAS ENGINES WITH WAUKESHA GAS ENGINES >350 CID/CYL

| MFG/Model | #CYL | CID/CYL | Continuous HP @ 1,200 rpm |
|---|---|---|---|
| **Caterpillar** | | | |
| G399 | 16 | 245 | 600 to 930 |
| G398 | 12 | 245 | 450 to 700 |
| G379 | 8 | 245 | 300 to 450 |
| G353 | 6 | 245 | 225 to 350 |
| G342 | 6 | 207 | 200 to 295 |
| **Waukesha** | | | |
| L7042 | 12 | 587 | 888 to 1,359 |
| L5790 | 12 | 482 | 726 to 1,114 |
| L5108 | 12 | 426 | 645 to 987 |
| F3521 | 6 | 587 | 432 to 674 |
| F2895 | 6 | 482 | 360 to 558 |

- Select a definition of greater than 240 CID/cyl
- Define affected gas engines as those greater than 350 CID/cyl or greater than or equal to 8 cylinders and greater than 240 CID/cyl

Both measures would essentially include only Caterpillar engines in the same power range as Waukesha. The second definition has a slight advantage over the first since it includes only Caterpillar engines that have Waukesha counterparts of about the same power (note that the greater-than-240-CID/cyl definition alone would include the Caterpillar G353, which has no large Waukesha counterpart). Therefore, the greater-than-350-CID/cyl or greater-than-or-equal-to-8 cylinders and greater-than-240-CID/cyl definition of affected gas engines is recommended.

With regard to one and two cylinder engines, it is recommended that they be excluded from potential standards of performance. This suggestion can be supported considering:

- At present these engines account for less than 10 percent of all gas engine horsepower and, therefore, are less significant $NO_x$ emitters than the larger gas engines used for oil and gas production
- These sources are numerous and widely dispersed in remote locations
- These engines are low rated* and therefore, probably have lower $NO_x$ emissions than the larger higher-rated gas engines

In addition to these factors, consideration should be given to the undeveloped control technology for these engines. A spokesman for one manufacturer noted that they are only currently preparing to measure $NO_x$ emissions from their one- and two-cylinder engines. Therefore, it is recommended that all one- and two-cylinder gas engines be exempted from potential standards of performance.

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*Operate at a small fraction of their potential power output.

In summary, then, it is recommended that the following criteria define gas engines that are to be affected by standards of performance:

- Affected facilities are defined as engines that are either greater than 350-CID/cyl or greater than 8-cylinder and greater than 240-CID/cyl
- All one or two cylinder gas engines are exempt from standards of performance

## 9.4 SELECTION OF BEST SYSTEM OF EMISSION REDUCTION

As discussed in Chapter 4, four emission control techniques, or combinations of these techniques, have been identified as demonstrated $NO_x$ emission reduction systems for stationary large-bore internal combustion engines. These techniques are: (1) retarded ignition or fuel injection, (2) air-to-fuel ratio changes, (3) manifold air cooling, and (4) derating power output (at constant speed). In general, all four techniques are applied by changing an engine operating adjustment. Manifold air cooling, however, may require a larger heat exchanger, and air-to-fuel changes may require turbocharger resizing.

These control techniques, described in Chapter 6, reduce $NO_x$ emissions primarily by lowering peak flame temperatures. Some of the techniques may result in increased fuel consumption and/or engine maintenance.

Fuel injection retard is the most effective $NO_x$ control technique for diesel-fueled engines, achieving maximum $NO_x$ reductions of about 65 percent. Similarly, air-to-fuel ratio change is the most effective $NO_x$ control technique for natural gas engines, achieving maximum $NO_x$ reductions of about 80 percent. Both retard and air-to-fuel ratio changes are effective in reducing $NO_x$ emissions from dual-fuel engines, achieving maximum $NO_x$ reductions of about 70 percent.

Other $NO_x$ emission control techniques exist but are not considered feasible alternatives. These techniques, also described in Chapter 6, include exhaust gas recirculation (EGR), combustion chamber modification (CCM), water induction, and catalytic reduction.

Exhaust gas recirculation tests have shown effective $NO_x$ reductions; however, the necessity for cooling the recirculated gas may lead to contamination of flow passages in the cooling heat exhanger as well as in the engine turbocharger and aftercooler. At present there is insufficient data on which to base conclusions and more development is required. Therefore, it is not considered a demonstrated emission control technique.

Data from smaller-bore diesel engines indicate that combustion chamber configuration has a significant effect on $NO_x$ emissions. However, none of the domestic large-bore engine manufacturers has thoroughly studied the effects of modified combustion chamber geometries on $NO_x$ emissions. Manufacturers have estimated that an extensive development program of three to five years would be required to establish the emission benefits of such a major engine redesign. The majority of the existing engines are primarily long-established designs that have been refined over the years to improve fuel economy and maintenance. Since there is insufficient data to draw conclusions, combustion chamber modification is not considered a demonstrated emission control technique.

The effect of water induction in large stationary internal combustion engines is similar to the effect in gas turbines. Significant $NO_x$ reductions are achieved due to the quenching effect of the presence of water. However, as discussed in Section 4.4.7, tests with water induction in large stationary internal combustion engines have shown unacceptable deposit build-up on the exhaust/intake valves from the use of untreated water, and severe lubricating oil contamination. Therefore, water induction is also not considered a demonstrated control technique.

Catalytic reduction of $NO_x$ in large stationary internal combustion engines is difficult to achieve and the capital cost could be high. Most large stationary internal combustion engines operate at air-to-fuel (i.e., mass flowrate [g/hr] of air into an engine divided by the mass flowrate of fuel [g/hr]) ratios that are typically much greater than stoichiometric and consequently the engine exhaust is characterized by high oxygen concentrations. Existing automotive catalytic converters, however, operate near stoichiometric conditions (i.e., low exhaust oxygen concentrations). These automobile catalysts are not effective in reducing $NO_x$ in the presence of high oxygen concentrations. Consequently, entirely different catalyst systems would be needed to reduce $NO_x$ emissions from large stationary internal combustion engines. Although such catalyst systems are currently under development and have been demonstrated for one very limited application (i.e., fuel-rich naturally aspirated gas engines), they have not been demonstrated for the broad range of IC engines manufactured, such as turbocharged engines, fuel-lean gas engines, or diesel engines. For these engines the reduction of $NO_x$ by ammonia injection over a precious metal (e.g., platinum) catalyst appears promising with $NO_x$ reductions of approximately 90 percent having been reported; however, the cost of such a system is high.

For a typical 1000 horsepower engine approximately 2 cubic feet of honeycomb catalyst (platinum based) would be required to ensure proper operation of the system. The cost of the catalyst was estimated at \$1500/cubic foot (in 1973). Assuming that the engine costs \$150/hp and that the cost of the catalyst accounts for about one-half the cost of the whole system (container, substrate, and catalyst), the capital investment for this control system represents approximately four percent of the engine purchase price.

The amount of ammonia required for an ammonia/catalyst $NO_x$ reduction system will depend on the $NO_x$ emission rate (g/hp-hr). Based on uncontrolled $NO_x$ emission rates of 9 to 22 g/hp-hr, and the cost of \$150/ton for the ammonia, the cost impact of injecting ammonia is approximately 5 to 15 percent of the total annual operating costs (\$/hp-hr) for natural gas engines. When this operating cost is combined with the capital cost of the catalytic system discussed above, the total cost increase is about 25 percent. Therefore, in continuous service applications this system is expensive compared to control techniques such as retard or air-to-fuel ratio changes.

It is also important to note that the consumption of ammonia can be expressed as a quantity of fuel since natural gas is generally used to produce ammonia. Assuming a conservative $NO_x$ emission rate of 20 g/hp-hr, and engine heat rate of 7500 Btu/hp-hr, a heating value of 21,800 Btu/lb for natural gas, and a requirement for approximately 900 lbs of gas per ton of ammonia produced, then the ammonia necessary for the catalytic reduction has the same effect on the supply of natural gas as a 2-percent increase in fuel consumption. Additional fuel is required to operate the plant which produces the ammonia.

Catalytic reduction, therefore, is currently not a demonstrated $NO_x$ emission control technique which could be used by all IC engines. Consequently, although catalytic reduction of $NO_x$ emissions could be used in a few isolated cases to comply with standards of performance, it could not be used as the basis for developing standards of performance which are applicable to all IC engines.

In summary, four emission control techniques have been identified as demonstrated $NO_x$ emission reduction systems for stationary large-bore internal combustion engines. These techniques are: (1) retarded ignition or fuel injection, (2) air-to-fuel ratio changes, (3) manifold air cooling, and (4) derating power output (at constant speed). Fuel injection retard is the most effective $NO_x$ control technique for diesel-fuel engines and air-to-fuel ratio change is the most effective $NO_x$ control technique for gas engines. Either technique is effective for dual-fuel engines.

The data and information presented in Chapters 4 and 6 clearly indicate that application of the control techniques mentioned above will reduce $NO_x$ emissions from internal combustion engines. It is not immediately clear, however, from this data and information whether the application of these emission control techniques to all IC engines would reduce $NO_x$ emissions to a specific level due to inherent differences in the uncontrolled emission characteristics of various engines. In general, engines with high uncontrolled $NO_x$ emission levels have relatively high controlled $NO_x$ emission levels and engines with low uncontrolled $NO_x$ emission levels have relatively low controlled $NO_x$ emission levels. To eliminate these inherent differences in $NO_x$ emission characteristics among various engines, the data were analyzed in terms of the degree of reduction in $NO_x$ emissions as a function of the degree of application of each emission control technique. Figures 4-27 and 4-31, reproduced here as Figures 9-13 and 9-14, illustrate the overall effectiveness of ignition retard and air-to-fuel ratio changes for reducing $NO_x$ emissions. For example, in Figure 9-13, the application of approximately eight of ignition retard results in about 40 percent reduction of $NO_x$ emissions. Thus, the data presented in Chapters 4 and 6 demonstrate that the same degree of application of each of these four $NO_x$ emission control techniques -- i.e., (1) retarded ignition or fuel injection, (2) air-to-fuel ratio changes, (3) manifold air cooling, and (4) derating power output (at constant speed) -- will result in essentially the same degree of reduction of $NO_x$ emissions from all large stationary internal combustion engines. Consequently, the ability to achieve certain percentage reductions in $NO_x$ emission levels is clearly demonstrated.

As can be seen from Figures 9-13 and 9-14, those included in

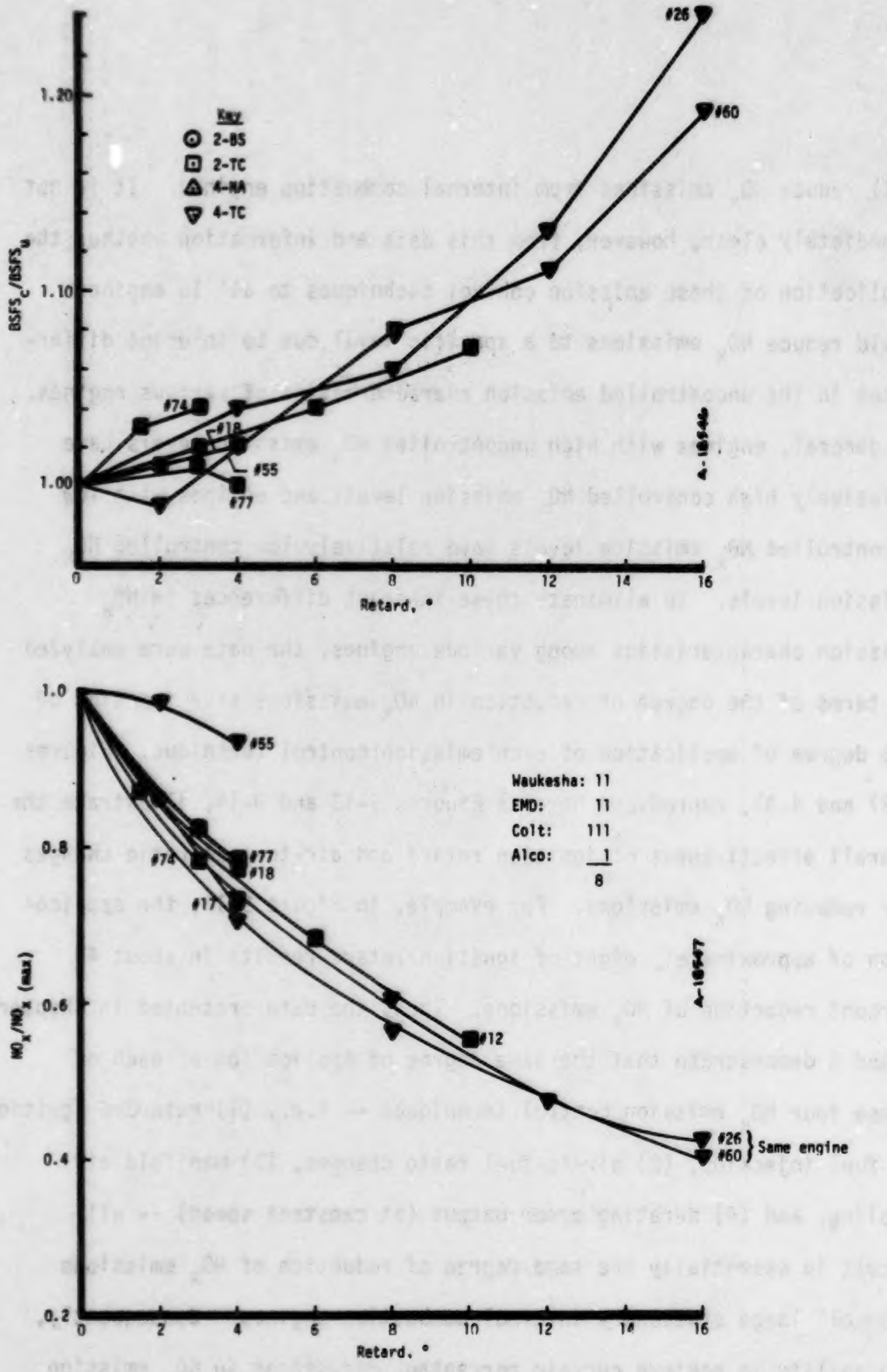

Figure 9-13. Effect of different amounts of retard on NO$_x$ emissions and fuel consumption for diesel engines.

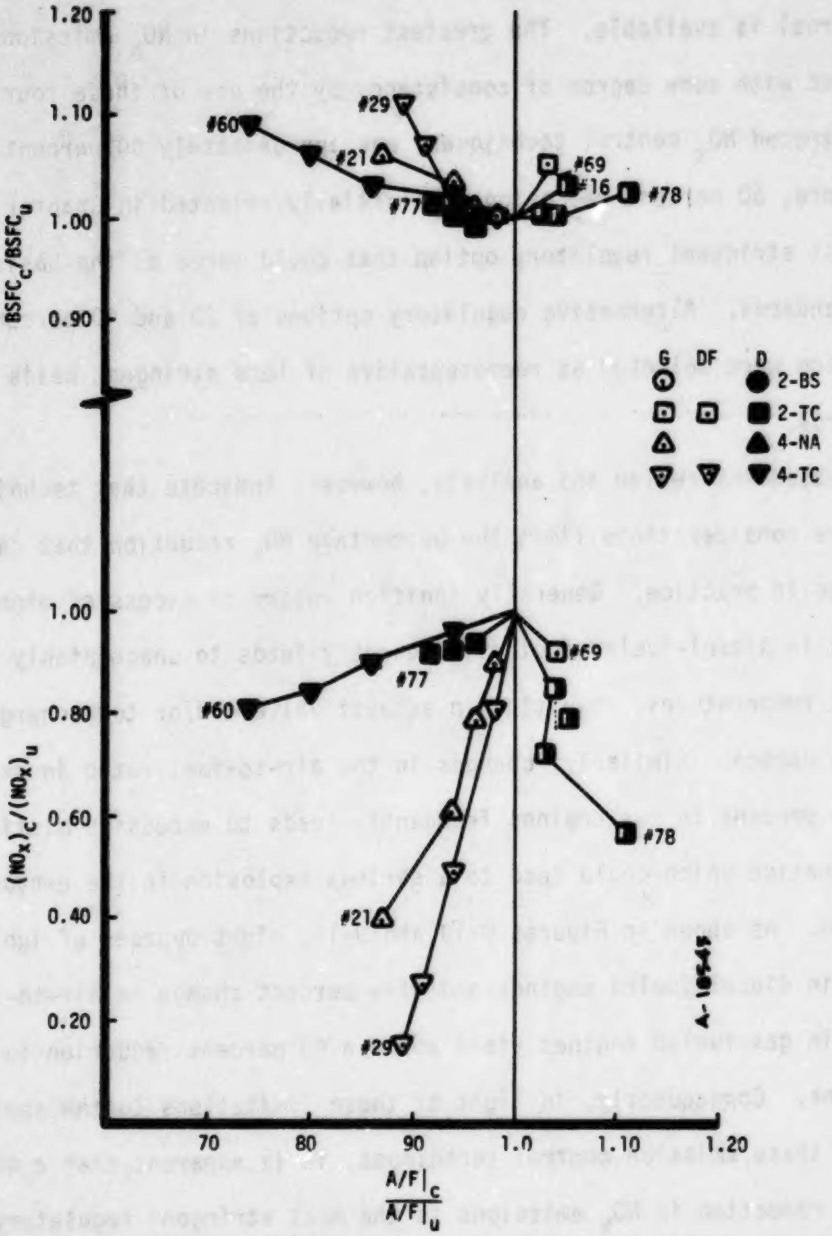

Figure 9-14. Effect of A/F changes on $NO_x$ emissions and fuel consumption.

Chapters 4 and 6, a wide range of regulatory options (i.e., basis for standards) is available. The greatest reductions in $NO_x$ emissions, achieved with some degree of consistency by the use of these four demonstrated $NO_x$ control techniques, was approximately 60 percent. Therefore, 60 percent reduction was initially selected in Chapter 6 as the most stringent regulatory option that could serve as the basis for the standards. Alternative regulatory options of 20 and 40 percent reduction were selected as representative of less stringent basis for standards.

Subsequent review and analysis, however, indicate that technical/hardware considerations limit the percentage $NO_x$ reduction that can be achieved in practice. Generally ignition retard in excess of eight degrees in diesel-fueled engines frequently leads to unacceptably high exhaust temperatures, resulting in exhaust valve and/or turbocharger turbine damage. Similarly, changes in the air-to-fuel ratio in excess of five percent in gas engines frequently leads to excessive misfiring or detonation which could lead to a serious explosion in the exhaust manifold. As shown in Figures 9-13 and 9-14, eight degrees of ignition retard in diesel-fueled engines and five percent change in air-to-fuel ratios in gas-fueled engines yield about a 40 percent reduction in $NO_x$ emissions. Consequently, in light of these limitations to the application of these emission control techniques, it is apparent that a 40-percent reduction in $NO_x$ emissions is the most stringent regulatory option which could be selected as the basis for standards of performance. An alternative of 20 percent $NO_x$ emission reductions was also considered a viable regulatory option which could serve as the basis for standards.

## Environmental Impacts

Standards of performance based on alternative I (20-percent reduction) would reduce national $NO_x$ emissions of about 14.6 million megagrams per year for all stationary sources by 72,500 megagrams annually in the fifth year after the standard went into effect. In contrast, standards of performance based on alternative II (40-percent reduction) would reduce national $NO_x$ emissions by about 145,000 megagrams annually in the fifth year after the standard went into effect. Thus, standards of performance based on alternative II would have a much greater impact on national $NO_x$ emissions than standards based on alternative I.

As discussed in Chapter 7, ambient air quality dispersion modeling, based on "worst case" conditions, indicates uncontrolled ambient air $NO_x$ levels near large stationary internal combustion engines can vary from approximately 60 percent of the National Ambient Air Quality Standard of 100 $\mu g/m^3$ to over twice the standard depending on the size of the engine. One calculation for a large gas engine yielded an uncontrolled ambient $NO_x$ level of about 220 $\mu g/m^3$.

These maximum concentrations, however, are located at distances extremely close to the source (0.3 km) because of the aerodynamic effects on plume rise as well as the relatively low height of the exhaust stack discharge. For example, it is estimated that at 1.0 km from the source, those concentrations would be only 15 percent of the above cited levels, well below the National Ambient Air Quality Standard.

In any event, standards of performance based on alternative I would reduce the highest calculated ambient air concentration of 220

$\mu g/m^3$ to about 180 $\mu g/m^3$, while standards based on alternative II would reduce this ambient $NO_x$ concentration level to about 100 $\mu g/m^3$. Thus, where ambient air $NO_x$ concentrations near large stationary internal combustion engines would be significant, standards of performance based on alternative II would be more effective in reducing ambient air $NO_x$ levels than standards of performance based on alternative I.

Standards of performance based on either alternative would, with the exception of naturally aspirated gas engines, not significantly effect carbon monoxide (CO) or hydrocarbon (HC) emissions from most engines. A typical diesel engine with a sales-weighted average uncontrolled CO emission level of approximately 2.9 g/hp-hr would experience an increase in CO emissions of about 0.75 g/hp-hr (26 percent) to comply with standards of performance based on alternative I, and an increase of about 1.5 g/hp-hr (52 percent) to comply with standards of performance based on alternative II. Total hydrocarbon emissions would increase a sales-weighted average uncontrolled emission level of 0.3 g/hp-hr by about 0.06 g/hp-hr (20 percent) to comply with standards based on alternative I, and would increase by about 0.1 g/hp-hr (33 percent) to comply with standards of performance based on alternative II.

Similarly, a typical dual-fuel engine with a sales-weighted average uncontrolled CO emission level of approximately 2.7 g/hp-hr would experience an increase in CO emissions of about 1.2 g/hp-hr (44 percent) and about 2.7 g/hp-hr (100 percent) to comply with standards of performance based on alternatives I and II, respectively. Total hydrocarbon emissions, however, would decrease by about 0.3 g/hp-hr (11 percent) from a sales-weighted average uncontrolled level of a approximatley 2.8 g/hp-hr to comply with standards of performance based on alternative I. To comply with standards of performance based on alternative II total hydrocarbon emissions would decrease 0.6 g/hp-hr (21 percent).

A typical turbocharged or blower scavenged gas engine with a sales-weighted average uncontrolled CO emission level of approximately 7.7 g/hp-hr would experience an increase in CO emissions of about 1.9 g/hp-hr (25 percent) to comply with standards of performance based on alternative I and about 3.8 g/hp-hr (49 percent) to comply with standards of performance based on alternative II. Total hydrocarbon emissions would increase a sales-weighted average uncontrolled level of approximately 1.8 g/hp-hr by about 0.2 g/hp-hr (11 percent) to comply with standards of performance based on alternative I. To comply with standards of performance based on alternative II total hydrocarbon emissions would increase by about 0.4 g/hp-hr (22) percent.

A typical naturally aspirated gas engine with a sales-weighted average uncontrolled CO emission level of approximately 7.7 g/hp-hr would experience an increase in CO emissions of about 3.9 g/hp-hr (51 percent) to comply with standards of performance based on alternative I and about 17 g/hp-hr (220 percent) to comply with standards of performance based on alternative II. Total hydrocarbon emissions would increase a sales-weighted average uncontrolled level of approximately 1.8 g/hp-hr by about 0.04 g/hp-hr (2 percent) to comply with standards of performance based on alternative I. To comply with standards of performance based on alternative II total hydrocarbon emissions would increase by about 0.08 g/hp-hr (4 percent).

The increase in ambient air CO levels due to compliance with $NO_x$ standards of performance based on either alternative would be small. For most engines, the application of standards of performance based on Alternative I would increase the maximum 8-hr ambient air CO concentration from about 0.23 $mg/m^3$ to 0.29 $mg/m^3$ for the typical diesel engine mentioned above. The application of standards of performance based on alternative II would increase the maximum 8-hour ambient air CO concentration to 0.35 $mg/m^3$. These values are insignificant compared to the National Ambient Air Quality Standard of 10 $mg/m^3$ for CO.

The application of standards of performance based on alternative I would increase the maximum 8-hr ambient air CO concentration from an uncontrolled concentration of 0.65 $mg/m^3$ to 0.94 $mg/m^3$ for the typical dual-fuel engine mentioned above. The application of standards of performance based on alternative II would increase the maximum 8-hr ambient air CO concentration to 1.3 $mg/m^3$.

For a typical turbocharged or blower scavenged gas engine, the increase in the maximum 8-hr ambient air CO concentration would be even less. The application of standards of performance based on alternative I would increase the maximum 8-hr ambient air CO concentration from an uncontrolled concentration of 0.15 $mg/m^3$ to 0.19 $mg/m^3$. The application of standards of performance based on alternative II would increase the maximum 8-hr ambient air CO concentration to 0.22 $mg/m^3$. These values are also insignificant compared to the National Ambient Air Quality Standard.

For a typical naturally aspirated gas engine, the application of standards of performance based on alternative I would increase the maximum 8-hr ambient air CO concentration from an uncontrolled concentration 0.77 mg/m$^3$ to 1.16 mg/m$^3$. The application of standards of performance based on alternative II would increase the maximum 8-hr ambient air CO concentration to approximately 2.0 mg/m$^3$.

Based on industry growth projections, an increase in national CO emissions of about 63,000 megagrams annually would be realized in the fifth year after the standards go into effect as a result of standards of performance based on alternative I. As a result of standards of performance based on alternative II an increase of about 216,000 megagrams annually would be realized in the fifth year after standards go into effect.

The large increase in CO emissions is due primarily to carbureted or naturally aspirated gas engines. These engines operate closer to stoichiometric conditions under which a small change in the air-to-fuel ratio results in a large increase in CO emissions. As shown in Figure 9-15, any significant $NO_x$ reduction is accompanied by a significant increase in CO.

As discussed earlier, though the total national CO emissions would increase significantly, ambient air CO concentrations in the immediate vicinity of these carbureted or naturally aspirated gas engines would not be adversely affected. As a result of the standards of performance based on alternative II, the maximum 1 hour ground level concentration from a typical engine would increase to about 2 mg/m$^3$ compared to the National Ambient Air Quality Standard of 10 mg/m$^3$. Ambient air $NO_x$

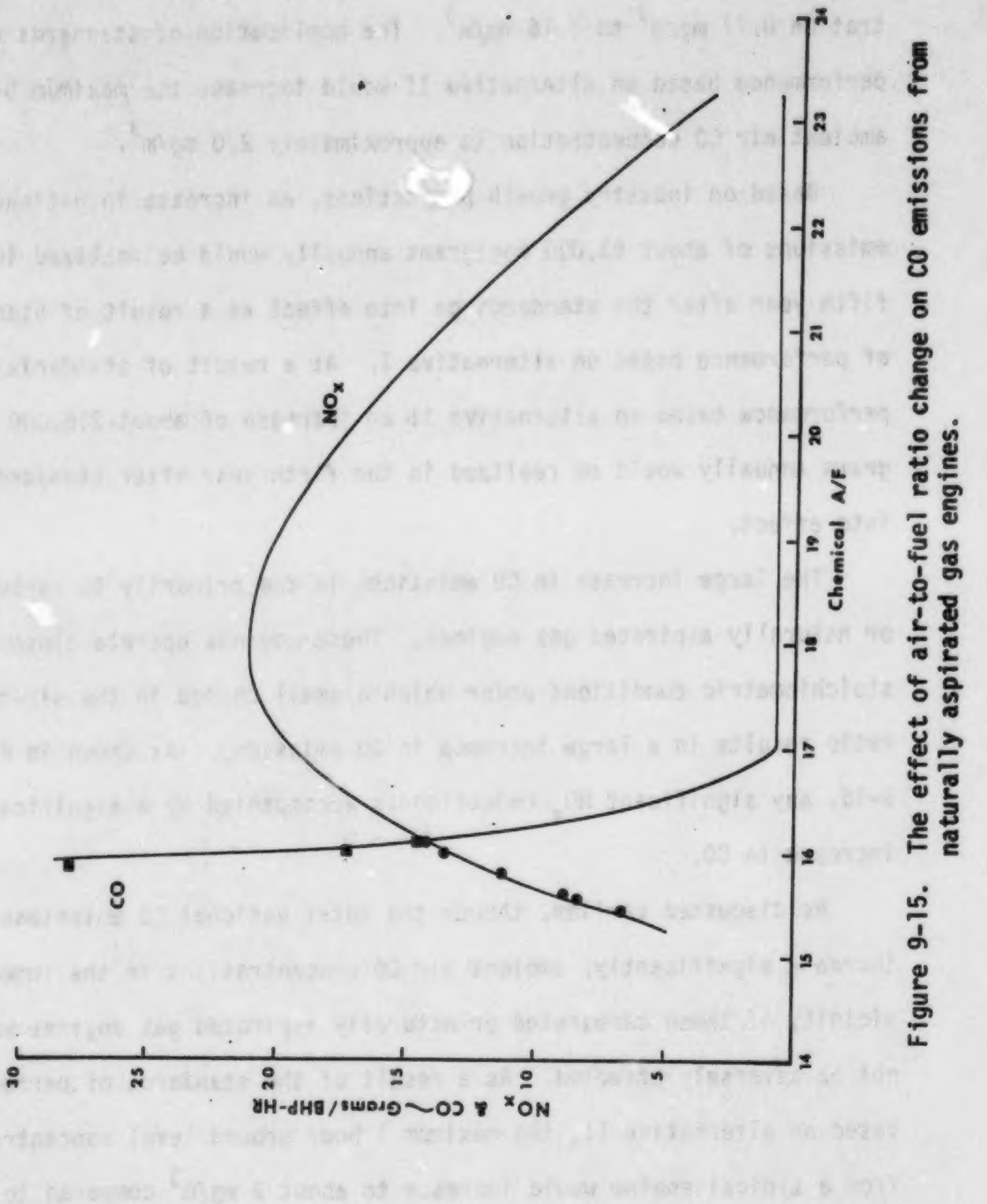

Figure 9-15. The effect of air-to-fuel ratio change on CO emissions from naturally aspirated gas engines.

concentrations from the same engine, however, would decrease concurrently about 40 percent, to a level less than half of the National Ambient Air Quality Standard of 100 $\mu g/m^3$.

Thus, there exists a trade-off between $NO_x$ emissions reduction and CO emissions increase, particularly for carbureted or naturally aspirated gas engines. EPA recognizes this trade-off and is concerned about it's attractiveness. It should be noted though that CO emissions are a local problem since they rapidly oxidize to $CO_2$. Additionally, most naturally aspirated gas engines are operated in remote locations where CO is not a problem. $NO_x$ emissions, however, are linked to the formation of photochemical oxidants and are subject to long range transport. $NO_x$ emissions reductions are also much harder to achieve than CO or HC emissions reductions which may be achieved more easily from other sources.

One alternative is to propose a CO emissions limit based on the use of oxidizing catalysts. These catalysts can provide CO and HC emissions reductions on the order of 90 percent. Initial capital costs are high, however, averaging about $7500 for a typical 1000 horsepower naturally aspirated gas engine or about 15 percent of the purchase price of the engine.

EPA feels these costs for control of CO emissions are unreasonable. The trade-off between $NO_x$ and CO emissions, however, seems reasonable. Therefore, CO was not selected for control by standards of performance.

Hydrocarbon emissions are also currently considered a pollutant of concern due to the impact on ambient air oxidant concentrations. However, although relationships are being developed to predict the HC emissions impact on oxidant concentrations, these concentrations depend on the ambient air $NO_x$/HC ratio. Thus, it is difficult to estimate the impact of increased HC emissions on ambient air oxidant concentrations. Furthermore, based on data in Appendix C.4, it is estimated that more than 90 percent of the total hydrocarbon emissions from gas engines and 75 percent of the total hydrocarbons from dual-fuel engines are methane, which is nonreactive and does not lead to oxidant formation. Finally, the increase in national total HC emissions based on either alternative, for most engines, is very small.

Standards of performance based on alternative I would increase national total HC emissions of 10.2 million megagrams by about 2,300 megagrams annually in the fifth year after the standards went into effect compared to an increase of about 4,600 megagrams annually associated with alternative II. Standards of performance based on alternative I would increase national reactive HC emissions by approximately 108 megagrams annually in the fifth year after the standards went into effect, compared to an increase of approximately 216 megagrams annually associated with alternative II.

As described in Chapter 3, stationary internal combustion engines are sources of $NO_x$, HC and CO emissions, with both $NO_x$ and HC contributing to oxidant formation. With regard to regulation of emission from IC engines, $NO_x$ emissions are of more concern than emissions of hydrocarbons for two reasons. First, $NO_x$ is emitted in greater quantities from stationary internal combustion engines than hydrocarbons. Second, a high priority has been assigned to development of standards of performance limiting $NO_x$ emissions from stationary sources are projected to increase by more than 40 percent in the 1975-to-1990 period. Applying best technology to all new sources would reduce this increase but would not prevent it from occurring. This unavoidable increase in $NO_x$ emissions is attributable largely to the fact that current $NO_x$ emission control techniques are based on combustion redesign. In addition, few $NO_x$ emission control techniques can achieve large (i.e., in the range of 90 percent) reductions in $NO_x$ emissions. In contrast, HC emissions are much easier to reduce. Large reductions from numerous sources are achievable with the installation of existing add-on control equipment. Consequently, EPA has assigned a high priority to the development of standards of performance for major $NO_x$ emission sources wherever significant reductions in $NO_x$ can be achieved.

The slight increase in HC emissions from IC engines associated with control of $NO_x$ from IC engines can be offset from other sources easier than $NO_x$ emissions can be reduced from other sources. Therefore, the adverse environmental impact of increased HC emissions due to the reduction in $NO_x$ emissions is considered small.

There would be essentially no water pollution impact of standards of performance based on either alternative I or alternative II. Only one control technique, increased manifold air cooling, could result in an additional discharge of water. However, most newly installed engines use a closed cooling system with no water discharge.

Standards of performance based on either alternative would also have no solid waste impact.

There would also be no adverse noise impact resulting from standards of performance based on either alternative. Fan noise levels from large-bore stationary engine installations could increase slightly as a result of increased cooling requirements; however, in typical installations such as municipal generator plants, pipeline compressor stations or industrial process plants, such increases are insignificant in comparison to existing noise levels.

Thus, as reflected in the summary Table 9-6, the environmental impacts of standards of performance based on either alternative are small and reasonable.

Energy Impacts

The potential energy impact of standards of performance based on either alternative is small. As discussed in Section 6.2, standards of performance based on alternative I could increase the fuel consumption of a typical blower scavenged or turbocharged gas engine by approximately one percent, whereas standards of performance based on alternative II could increase the fuel consumption by approximately two percent. A typical 2000 horsepower blower scavenged or turbocharged gas engine has an uncontrolled fuel consumption of approximately 343,000 scf natural gas per day (the energy equivalent of 2842 gallons oil per day). Increases translate into actual increased fuel consumption of approximately 3500 scf and 7000 scf natural gas per day (the energy equivalent of 25 and 50 gallons oil per day).

Standards of performance based on alternative I would increase the fuel consumption of a typical naturally aspirated gas engine by approximately six percent. Standards of performance based on alternative II, however, would increase the fuel consumption by approximately eight percent.

A typical 2000 horsepower naturally aspirated gas engine has an

Table 9-6 ENVIRONMENTAL IMPACTS OF ALTERNATIVES

| Pollutant | Base Level[a] | Alternative I | Alternative II |
|---|---|---|---|
| National $NO_x$ Emissions | $14.6 \times 10^6$ megagrams | Reduced by 72,500 megagrams annually 5 years after standard goes into effect | Reduced by 145,000 megagrams annually 5 years after standard goes into effect |
| National CO Emissions | $33.0 \times 10^6$ megagrams | Increased by 63,000 megagrams annually 5 years after standard goes into effect | Increased by 216,000 megagrams annually 5 years after standard goes into effect |
| National Total HC Emissions | $10.2 \times 10^6$ megagrams | Total Hydrocarbons<br>Increased by 2,300 megagrams annually 5 years after standard goes into effect<br>Reactive Hydrocarbons<br>Increased by 108 megagrams annually 5 years after standard goes into effect | Total Hydrocarbons<br>Increased by 4,600 megagrams annually 5 years after standard goes into effect<br>Reactive Hydrocarbons<br>Increased by 216 megagrams annually 5 years after standard goes into effect |
| Water Pollution | -- | No increase | No increase |
| Solid Waste | -- | No increase | No increase |
| Noise | -- | No adverse impact | No adverse impact |

[a] Total U.S. emission from stationary sources as per EPA Nationwide Air Pollutant Inventory for 1975

uncontrolled fuel consumption of approximately 343,000 scf natural gas per day (the energy equivalent of 2842 gallons oil per day).

These percentage increases represent actual fuel consumption increases of about 20,600 and 27,400 scf natural gas per day (the energy equivalent of 142 and 189 gallons oil per day).

Standards of performance based on alternative I could increase the fuel consumption of a typical dual-fuel engine by approximately one percent, whereas standards of performance based on alternative II could increase the fuel consumption by approximately three percent. A typical dual-fuel engine rated at 2000 horsepower has an uncontrolled fuel consumption of approximately 297,000 scf natural gas per day (the energy equivalent of 2150 gallons oil per day). These percentage increases represent an actual fuel consumption increase of about 6100 and 12,200 scf natural gas per day (the energy equivalent of 43 and 86 gallons oil per day).

Standards of performance based on alternative I could increase the fuel consumption of a typical diesel engine by approximately three percent, whereas standards of performance based on alternative II could increase the fuel consumption by approximately seven percent. For a typical diesel engine rated at 2000 horsepower, with an uncontrolled fuel consumption of approximately 2320 gallons of oil per day, these percentage increases represent actual fuel consumption increases of about 70 and 160 gallons of oil per day.

Thus, the potential energy impact in the fifth year after the standard goes into effect, based on alternative I, would be equivalent to approximately 1.03 million barrels of oil per year compared to an uncontrolled fuel consumption of IC engines affected by the standard of 31 million barrels per year. The potential energy impact in the fifth year after the standard goes into effect, based on alternative II, would be equivalent to approximately 1.5 million barrels of oil per year.

It should be noted that the largest increase represents only 0.01 percent of the 1977 domestic consumption of crude oil and natural gas. The largest increase also represents only 0.03 percent of the projected total oil imported to the United States five years after the standards go into effect.

Thus, as reflected in the summary Table 9-7, the energy impacts of standards of performance based on either alternative are small and reasonable.

TABLE 9-7. ENERGY IMPACTS OF ALTERNATIVES

| Engine Fuel Type | Uncontrolled[a] Fuel Consumption (gal/day) | Increase in Fuel Consumption (gal/day) | |
|---|---|---|---|
| | | Alternative I | Alternative II |
| Gas[b] | 2842[d] | 25 | 50 |
| Gas[c] | 2842 | 142 | 189 |
| Dual Fuel | 2151 | 43 | 86 |
| Diesel | 2317 | 70 | 162 |
| Totals of all new engines after 5 years | 31,000,000 bbls oil/yr | 1.03 million bbls oil/yr | 1.5 million bbls oil/yr |

[a] Typical 2000 horsepower engine
[b] Blower scavenged or turbocharged
[c] Naturally aspirated
[d] Expressed as equivalent oil consumption

Economic Impact of Alternatives

Manufacturers of stationary internal combustion engines would incur additional costs due to standards of performance. These costs however would be small. As discussed in Section 6.3, these costs are a result of one or more of the following activities that may be needed to manufacture engines which meet standards of performance: (1) extended testing to verify the effectiveness of a particular control approach; (2) development and application of $NO_x$ controls to existing engine designs; and (3) engineering, tooling and pattern costs for minor redesign of an engine family to accomodate application of a control technique. It is estimated that the total costs to the manufacturer for each engine model family, including development, durability tests, and retooling, would be approximately: (1) $125,000 for retard and air-to-fuel change; (2) $150,000 for manifold air temperature reduction; and (3) $25,000 for derate. For each manufacturer, therefore, total costs would vary depending on (1) the number of engine model families produced, (2) their degree of advancement in emission testing, (3) the uncontrolled emission levels of their engines, (4) the development and durability testing required to produce engines that can meet proposed standards of performance, and (5) the emission control technique selected.

As reported in Section 8.4, the economic impacts on manufacturers arising from these cost penalties associated with standards of performance based on either alternative would be small.

The manufacturer's total capital investment requirements for developmental testing of engine models is estimated to be about $4.5 million to comply with standards of performance based on alternative I and about $5 million to comply with standards of performance based on alternative II. These expenditures would be made over a two year period. Analysis of the financial statements of engine manufacturers or their parent companies indicates that the manufacturers' overhead budgets are sufficient to support the development of these controls without adverse impact on their financial position.

As discussed in Section 8.4.1.2, manufacturers would not experience significant differential cost impacts among competing engine model families. The cost penalties summarized in Table 8-16 reflect the range of total annualized cost penalties to the end use applications for each engine fuel type and manufacture for each alternative. These costs for each major end use, the cross-price inelasticities, and the importance of each end use market to the companies' total internal combustion sales were analyzed to determine the relative sales advantages between companies. Consequently, these analyses indicated that no significant sales advantages or disadvantages would develop among competing manufacturers for standards of performance based on either alternative. Based on "worst case" assumptions the maximum intra-industry sales losses would be about six percent as a result of standards of performance based on either alternative. Thus, the intra-industry impacts would be moderate and not cause any major dislocations within the industry.

These total annualized cost penalties imposed on IC engines by standards of performance would also have very little impact with regard to increasing sales of gas turbines. Turbines do not compete with internal combustion engines based on annualized costs alone, due to their higher operating costs (fuel). As discussed in Subsection 8.4.1.3, the total annualized cost penalty associated with standards of performance based on either alternative would bring the cost of owning internal combustion engines up to that of turbines in only one case -- diesel internal combustion engines used in electric generation. This conclusion, however, is based on limited data. It is quite likely, however, that this manufacturer's line of diesel engines, through minor combustion modifications, could reduce their $NO_x$ emission to levels comparable to that of other manufacturers. Further, due to technical limitations, economic considerations, and customer preference, it is unlikely that IC engine users will switch to gas turbines. For example, it is unlikely that turbines would replace diesel engines in plants using banks of smaller engines, unless the entire bank were replaced with one turbine.

Standards of performance based on alternative I would result in no loss of sales to gas turbines whereas standards of performance based on alternative II would result in the possible loss of sales for one diesel manufacturer.

Thus, the economic impacts on the manufacturers arising from standards of performance based on either alternative are considered small and reasonable.

The application of $NO_x$ controls will also increase costs to the engine user. The magnitude of this increase will depend upon the amount and type of emission control applied. As was shown in Section 6.3, various control approaches affect initial costs, fuel consumption, and maintenance differently. Fuel penalites, though, are the major factor affecting this increase for high usage engines.

The following four end uses were selected to represent the major applications of diesel, dual-fuel, and natural gas engines: (1) diesel engine, electrical generation; (2) dual-fuel engine, electrical generation, (3) gas engine, oil and gas transmission and (4) gas engine, oil and gas production.

The total annualized cost of a typical uncontrolled diesel fuel, electrical generation engine is about 2.5¢/hp-hr. For a typical 2000 horsepower engine operating 8000 hours per year, this total annualized cost would be about $450,000 per year. Standards of performance based on alternative I would increase this total annualized cost by about 0.04¢/hp-hr (1.5 percent). Similarly, standards of performance based on alternative II would increase the total annualized cost by about 0.11¢/hp-hr (4.5 percent). For the engine mentioned above, these values translate into dollar amounts of about $6,400 and $17,600 respectively.

The total annualized cost of a typical uncontrolled dual-fuel, electrical generation engine is about 2.8¢/hp-hr. For a typical 2000 horsepower engine operating 8000 hours per year this would be about $448,000 per year. As a result of standards of performance based on alternative I this total annualized cost would increase by about 0.07¢/hp-hr (2.5 percent). As a result of standards of performance based on alternative II, this total annualized cost would increase by about 0.09¢/hp-hr (3.2 percent). For this engine these values translate into dollar amounts of about $11,200 and $14,400 respectively.

The total annualized cost of a typical uncontrolled natural gas fuel, oil and gas transmission engine is about 2.2¢/hp-hr. For a typical 2000 horsepower engine operating 8000 hours per year, this total annualized cost would be about $354,000 per year. Standards of performance based on alternative I would increase this total annualized cost by about 0.02¢/hp-hr (1 percent). Standards of performance based on alternative II would increase the total annualized cost by about 0.04¢/hp-hr (2 percent). For the engine mentioned above, these values translate into dollar amounts of about $3500 and $7100, respectively.

The total annualized cost of a typical uncontrolled natural gas fuel, oil and gas production engine is about 2.2¢/hp-hr. For a typical 2000 horsepower engine operating 8000 hours per year, this total annualized cost would be about $250,000 per year. Standards of performance based on alternative I would increase this total annualized cost by about 0.14¢/hp-hr (6.1 percent). Standards of performance based on alternative II would increase this total annualized cost by about 0.16¢/hp-hr (8 percent). For the typical engine mentioned above, these values translate into dollar amounts of about $22,400 and $25,600, respectively.

Total uncontrolled annualized costs of $580 million by all large stationary internal combustion engine users would increase by about $25 million to comply with standards of performance based on alternative I and by about $32 million to comply with standards of performance based on alternative II in the fifth year after standards go into effect.

The manufacturer's capital test requirements would be regarded as an added expense and most likely passed on to the engine users in the form of higher prices. Therefore, users of internal combustion engines would have to expend additional capital to purchase more expensive engines. This capital cost penalty however, is small. A two percent increase in engine price would be expected on the average as a result of standards of performance based on either alternative. Typical initial costs for uncontrolled diesel and dual-fuel, electrical generation engines, and natural gas, oil and gas transmission engines are about $150/hp-hr. Initial costs for natural gas fuel, oil and gas production engines are about $50/hp. For typical 2000 horsepower engines, these initial capital costs would be about $300,000 and $100,000, respectively.

Standards of performance based on either alternative would increase the initial capital cost of a typical diesel or dual-fuel, electrical generation engine or natural gas fuel, oil and gas transmission engine rated at 2000 horsepower by about $6000.

In contrast, standards of performance based on either alternative would increase the initial capital cost of a typical natural gas, oil and gas production engine rated at 2000 horsepower by about $2000.

The total additional capital cost for all users would equal about $9.6 million per year on a cumulative basis on either alternative compared to total uncontrolled costs of all new engines of $450 million per year.

As discussed in Section 8.4, the economic impacts on users arising from the cost penalties associated with standards of performance based on either alternative would be small. In general, these impacts translate into price increases for the end products or services provided by the industrial and commercial users of large stationary internal combustion engines. The electric utility industry would realize a price increase after five years of 0.02 percent to comply with standards of performance based on either alternative. After five years, delivered natural gas prices would increase 0.02 percent due to the application standards of performance based on alternative I and 0.04 percent due to the application standards of performance based on alternative II.

Even after a full phase-in period of 30 years, during which new controlled engines would replace all existing uncontrolled engines, the electric utility industry would realize a price increase of only 0.1 percent to comply with standards of performance based on either alternative. Similarly, delivered natural gas prices would increase only 0.1 percent due to the application of standards of performance based on alternative I and 0.3 percent to comply with standards of performance based on alternative II. Thus, the economic impacts of standards of performance based on either alternative are considered small and reasonable.

## Conclusions

Based on this assessment of the impacts of each alternative, and given the fact that alternative II achieves a greater degree of $NO_x$ reduction, it is selected as the best technological system of continuous emission reduction of $NO_x$ from stationary large-bore IC engines considering the cost of achieving such emission reduction, any nonair quality health and environmental impact and energy requirements.

Table 9-8, which follows, summarizes the economic impacts of each alternative.

TABLE 9-8 ECONOMIC IMPACTS OF ALTERNATIVES

| Impact | Uncontrolled Level of Cost | Alternative I | Alternative II |
|---|---|---|---|
| Impact on Manufacturer | | | |
| Capital budget requirements | -- | $4.5 million over two years; able to generate internally from profits. | $5 million over two years; able to generate internally from profits. |
| Intra-industry competition | -- | Maximum sales loss unlikely to exceed 6% of internal combustion engine sales for any firm. | 6% maximum loss for any firm |
| Competition from gas turbines | -- | No losses. | Possible sales loss for one diesel manufacturer. |
| Impact on End-Use Applications | | | |
| Total annualized cost[a] | | | |
| Diesel fuel, electrical generation | 2.5¢/hp-hr | Base increased by 0.04¢/hp-hr | Increased by 0.11¢/hp-hr |
| Dual-fuel, electrical generation | 2.8¢/hp-hr | Increased by 0.07¢/hp-hr | Increased by 0.09¢/hp-hr |
| Natural gas fuel, oil and gas transmission | 2.2¢/hp-hr | Increased by 0.02¢/hp-hr | Increased by 0.04¢/hp-hr |
| Natural gas fuel, oil and gas production | 2.2¢/hp-hr | Increased by 0.14¢/hp-hr | Increased by 0.16¢/hp-hr |
| Totals of all new engines after 5 years | $580 million | Increased by $25 million | Increased by $32 million |
| Capital Cost Penalty[a] | | | |
| Diesel fuel, electrical generation or dual fuel, electrical generation or natural gas fuel, oil and gas transmission | $150/hp | Increased by $3.00/hp | Increased by $3.00/hp |
| Natural gas fuel, oil and gas production | $ 50/hp | Increased by $1.00/hp | Increased by $1.00/hp |
| Totals etc. | $450 million | $9.6 million on a cumulative basis over first 5 years after standards go into effect. | $9.6 million on a cumulative basis over first 5 years after standards go into effect. |
| Impact on Product Prices and Users | | | |
| Electricity prices | -- | U.S. electric bill up 0.02% after 5 years. U.S. electric bill up 0.1% after full phase-in. | U.S. electric bill up 0.02% after 5 years. U.S. electric bill up 0.1% after full phase-in[b]. |
| Gas prices | -- | Delivered natural gas prices up 0.02% after 5 years. Delivered natural gas prices up 0.1% after full phasein. | Delivered natural gas prices up 0.04% after 5 years. Delivered natural gas prices up 0.3% full phasein. |

[a] Assumed typical 2000 horsepower engine operating 8000 hours per year in all cases

[b] Full phase-in implies replacement of all existing engines

## 9.5 SELECTION OF FORMAT FOR THE STANDARDS

A number of different formats could be used to limit $NO_x$ emissions from large stationary internal combustion engines. Standards could be developed to limit emissions in terms of: (1) percent reduction, (2) mass emission per unit of energy (power) output, (3) mass emissions per unit of energy (fuel) input, or (4) concentration of emissions in the exhaust gases discharged to the atmosphere.

Analysis of the effectiveness of the various demonstrated $NO_x$ emission control techniques clearly shows that what is demonstrated is the ability to achieve a percent reduction in $NO_x$ emissions. In other words, application of each emission control technique to the same degree (i.e., eight degrees of ignition retard to five percent change in air-to-fuel ratios) will result in essentially the same percentage reduction in $NO_x$ emissions. However, a percent reduction format is highly impractical for two reasons. First a reference uncontrolled $NO_x$ emission level would have to be established for each manufacturer's engine, a difficult task since some manufacturers produce as many as 25 models which are sold with several ratings. Second, a reference uncontrolled $NO_x$ emission level would have to be established for any new engines developed after promulgation of the standard. This would be quite simple for engines that employed $NO_x$ control techniques such as ignition retard or air-to-fuel ratio change to comply with standards. Emissions could be measured without the use of these techniques. For engines designed to comply with the standards through the use of combustion chamber modification, however, this would not be possible. Thus, new engines would receive no credit for the $NO_x$ emission reduction achieved by combustion chamber redesign.

A mass-per-unit-of-energy-output format, typically referred to as brake-specific emissions (g/hp-hr), relates the total mass of $NO_x$ emissions to the engine's productivity. Although brake-specific mass standards (g/hp-hr) appear meaningful because they relate directly to the quantity of emissions discharges into the atmosphere, there are disadvantages in that enforcement of mass standards would be costly and complicated in practice. This can be illustrated by examining the relationship between brake-specific mass emissions ($BSNO_x$) and the parameters that would have to be measured in the field:

$$BSNO_x \sim NO_x\ (m_e)\ (1/w)$$

where:

$BSNO_x$ = grams $NO_x$/horsepower-hour

$NO_x$ = concentration of $NO_x$ in exhaust, parts per million (ppm)

$m_e$ = exhaust mass flowrate, lb/hr

w = power output, horsepower

Thus, exhaust flow and power output would have to be determined in addition to $NO_x$ concentration. For example, to determine exhaust gas flowrate, one of three methods can be used: (1) directly measure exhaust volume flowrate: (2) measure inlet air and fuel flowrate; and (3) measure all exhaust carbon constituents (primarily HC, CO, $CO_2$), and fuel flow, and conduct a fuel analysis.

Since large internal combustion engines have very large exhaust flowrates (in excess of 50,000 cfm), exhaust flowrates are difficult to determine accurately in either the field or laboratory. Similarly, the accurate measurement of inlet airflow is difficult. Thus, methods (1) or (2) are unlikely to be used in practice for large engines.

Although method (3) has been used in the field to determine brake-specific mass emissions, the measurement of three additional exhaust gas constituents and the fuel analysis considerably complicate the test procedure(31). Moreover, it is difficult to accurately measure fuel flow over a short interval (typically less than 1/2 hour, which would be required for a performance test). Thus, the determination of exhaust gas flowrates in the field is difficult and complicated.

Another disadvantage of the brake-specific mass emissions format is that power output must be determined. Power can be determined from an engine dynamometer in the laboratory, but dynamometers cannot be used in the field. Power output could be determined by: (1) inferring the power from engine operating parameters (fuel flow, rpm, manifold pressure, etc.) or (2) inferring engine power from the output of the generator or compressor attached to the engine. In practice, however, these approaches are time consuming and are less accurate than dynamometer measurements.

A format limiting $NO_x$ emissions per unit of energy (fuel) input would be specified in terms of grams $NO_x$ per joule fuel input (equivalent to lb $NO_x$/M Btu). The advantage of this format is that no power measurement would be required, thereby simplifying enforcement. However, as with a brake-specific mass emission format, total exhaust gas mass flowrates must be calculated, and as was discussed earlier, all methods for this determination are difficult under field conditions.

In addition, standards of performance based on fuel input could penalize more efficient engines, which typically operate at higher temperatures and pressures, leading to higher $NO_x$ emissions. For example, given two engines with the same brake-specific emissions, the more efficient engine, which consumes less fuel, will have a higher fuel-based emissions level because the ratio of mass $NO_x$ emissions per unit of fuel energy input has a smaller denominator. Thus, this format could offset other incentives manufacturers have to develop more efficient engines.

Another possible format would be to limit the concentration of $NO_x$ emissions in the exhaust gases discharged to the atmosphere. Concentrations would be specifed in terms of parts of $NO_x$ per million (ppm) parts of exhaust (volumetric). The major advantage of this format is in the simplicity of its enforcement. As compared to the formats discussed previously, only a minimum of data and calculations are required, which decreases testing costs and minimizes errors in determining compliance with an emission standard. Measurements are direct; only $NO_x$ and $O_2$ concentration measurements of exhaust gas must be made. The $NO_x$ measurement reads out directly in ppm of dry exhaust, and the oxygen measurement, required to prevent a user from diluting the exhaust gas with air and lowering the $NO_x$ concentration, reads out in percent-$O_2$. A reference concentration of oxygen, however, must be established for this format.

The primary disadvantages associated with concentration standards are: (1) a standard could be circumvented by dilution of exhaust gases discharged into the atmosphere, which lowers the concentration of the emissions but does not reduce the total mass emitted, and (2) a concentration standard could penalize high efficiency engines because more efficient engines generally discharge higher concentrations of $NO_x$

emissions due to higher operating temperatures and pressures as mentioned above (although the mass emission rate may be the same as a lower efficiency engine). A concentration standard based on low efficiency engines, therefore, could penalize high efficiency engines. Both these problems, however, can be overcome through the use of appropriate "correction" factors.

Since the percent reduction format is impractical, and the problems associated with the enforcement of mass standards (mass per unit energy output or input) appear to outweigh the benefits, the concentration formation was selected as the format for standards of performance for large stationary internal combustion engines.

As mentioned above, because a concentration standard can be circumvented by dilution of the exhaust gases, measured concentrations must be expressed relative to some fixed dilution level. For combustion processes, this can be accomplished by correcting measured concentrations to a reference concentration of oxygen. The oxygen concentration in the exhaust gases is related to the excess (or dilution) air. Typical oxygen concentrations in large-bore internal combustion engines can range from 8 to 16 percent but are normally about 15 percent. Thus, referencing the standard to a typical level of 15 percent oxygen would prevent circumvention by dilution. (Section 9.6 discusses the correction factor for adjusting measured $NO_x$ concentrations to an oxygen concentration of 15 percent).

As also mentioned above, selection of a concentration format could penalize high efficiency internal combustion engines. These highly efficient engines generally operate at higher temperatures and pressures, and as a result discharge gases with higher $NO_x$ concentrations than less efficient engines, although both engines' brake-specific mass emissions could be the same. Thus, a concentration standard based on low efficiency engines could effectively require more stringent controls for high efficiency engines. Conversely, a concentration standard based on high efficiency engines would require no controls. Consequently, selecting a concentration format for standards of performance requires an efficiency factor to permit higher $NO_x$ emission from more efficient engines.

The incentive for manufacturers to increase engine efficiency is to lower engine fuel consumption. Therefore, the objective of an efficiency adjustment factor should be to give an emissions credit for the lower fuel consumption of more efficient internal combustion engines. Since the fuel consumption of internal combustion engines varies linearly with efficiency, a linear adjustment factor is selected to permit increased $NO_x$ emissions from highly efficient internal combustion engines. A linear efficiency adjustment factor also effectively limits $NO_x$ emissions to a constant mass emission rate per unit of power output.

The efficiency adjustment factor needs to be referenced to a baseline efficiency. Most large existing stationary internal combustion engines fall in the range of 30 to 40 percent efficiency. Therefore, 35 percent is selected as the baseline efficiency. The efficiency of internal combustion engines is usually expressed in terms of heat rate. The heat rate of engines operating at 35 percent efficiency is about 7270 Btu/hp-hr. Thus, the following linear adjustment factor is selected to permit increased $NO_x$ emissions from high efficiency large stationary internal combustion engines:

$$X_a = X \frac{7270}{Y}$$

where:

$X_a$ = Adjusted $NO_x$ emissions permitted at 15 percent oxygen, ppmv

$X$ = $NO_x$ emission limit specified in the standards at 15 percent oxygen on a dry basis

$Y$ = LHV heat input per unit of power output (Btu/hp-hr)

NOTE: Above adjustment is made at standard atmospheric conditions of 29.92 m Hg, 85°F, and 75 grains moisture per pound of dry air.

This efficiency adjustment factor permits a linear increase in $NO_x$ emission with increased efficiencies above 35 percent. This adjustment would not be used to adjust the emission limit downward for internal combustion engines with efficiencies of less than 35 percent. This efficiency adjustment factor also applies only to the IC engine itself and not the entire system of which the engine may be a part. Since Section 111 of the Clean Air Act requires the use of the best system of emission reduction in all cases, this precludes the application of the efficiency adjustment factor to an entire system. For example, IC engines with waste heat recovery may have a higher overall efficiency than the IC engine alone. Thus, the application of the efficiency adjustment factor to the entire system would permit greater $NO_x$ emissions because of the system's higher overall efficiency, and would not necessarily require the use of the best demonstrated system of emission reduction on the IC engine.

## 9.6 SELECTION OF NUMERICAL EMISSION LIMITS

### Overall Approach

As mentioned earlier, it is difficult to select a specific $NO_x$ emission limit which all IC engines could meet primarily through the use of ignition retard or air-to-fuel ratio change. Because of inherent differences among various IC engines with regard to uncontrolled $NO_x$ emission levels, there exists a rather large variation within the data and information included in the SSEIS concerning controlled $NO_x$ emission levels. Generally speaking, engines with relatively low uncontrolled $NO_x$ emissions achieved low controlled $NO_x$ emission levels, and engines with high uncontrolled $NO_x$ emission levels achieved relatively high controlled $NO_x$ emission levels. Consequently, the following alternatives were considered for selection of the numerical concentration emission limits based on a 40 percent reduction in $NO_x$ emissions:

1. Apply the 40 percent reduction to the highest observed uncontrolled $NO_x$ emission level.
2. Apply the 40 percent reduction to a sales-weighted average uncontrolled $NO_x$ emission level.
3. Apply the 40 percent reduction to this sales-weighted average uncontrolled $NO_x$ emission level plus one standard deviation.

The highest observed uncontrolled $NO_x$ emission levels for diesel, dual-fuel and gas engines discussed in alternative I above can be found in Figures 4-49(a-c), respectively. The uncontrolled levels for dual-fuel engines are generally lower than those for diesels, which are generally lower than those for gas engines. The highest uncontrolled levels for each fuel type are as follows: (1) gas, 29 g/hp-hr, (2) dual-fuel, 15.0 g/hp-hr, and (3) diesel, 19 g/hp-hr.

The sales-weighted uncontrolled $NO_x$ emission levels which are used as the base levels in the second alternative are discussed in Section 4.3.4. It was noted that uncontrolled $NO_x$ emission levels vary among both engines of the same fuel type and of different fuels (even after considering the effects of ambient conditions and measurement methods). Therefore, an average uncontrolled level was determined by applying a sales-weighting to each manufacturer's average uncontrolled $NO_x$ emissions for engines of each fuel type (see Section 4.3.4). The sales-weighting, based on horsepower sold, gives more weight to those engine models which have the highest sales. The sales-weighted average uncontrolled $NO_x$ emission level for each engine fuel type are as follows: (1) gas, 15 g/hp-hr, (2) dual-fuel, 8 g/hp-hr, and (3) diesel, 11 g/hp-hr.

The third alternative incorporates a "margin for engine variability" by adding one standard deviation to the sales-weighted average uncontrolled $NO_x$ emission level and then applying the 40 percent reduction. Standard deviations discussed were calculated from the uncontrolled $NO_x$ emission data included in the SSEIS, assuming it had a normal distribution. A subsequent statistical evaluation of the data indicated that this assumption was valid (see Appendix C for a complete discussion). The standard deviations for each engine fuel type are as follows: (1) gas, 4 g/hp-hr, (2) dual-fuel, 3.2 g/hp-hr and (3) diesel, 3.7 g/hp-hr.

The standard deviation of the uncontrolled $NO_x$ emission data base is relatively large compared to the sales-weighted average uncontrolled internal combustion engines:

$$X_a = X \frac{7270}{Y}$$

where:

$X_a$ = Adjusted $NO_x$ emissions permitted at 15 percent oxygen, ppmv

$X$ = $NO_x$ emission limit specified in the standards at 15 percent oxygen on a dry basis

$Y$ = LHV heat input per unit of power output (Btu/hp-hr)

NOTE: Above adjustment is made at standard atmospheric conditions of 29.92 m Hg, 85°F, and 75 grains moisture per pound of dry air.

This efficiency adjustment factor permits a linear increase in $NO_x$ emission with increased efficiencies above 35 percent. This adjustment would not be used to adjust the emission limit downward for internal combustion engines with efficiencies of less than 35 percent. This efficiency adjustment factor also applies only to the IC engine itself and not the entire system of which the engine may be a part. Since Section 111 of the Clean Air Act requires the use of the best system of emission reduction in all cases, this precludes the application of the efficiency adjustment factor to an entire system. For example, IC engines with waste heat recovery may have a higher overall efficiency than the IC engine alone. Thus, the application of the efficiency adjustment factor to the entire system would permit greater $NO_x$ emissions because of the system's higher overall efficiency, and would not necessarily require the use of the best demonstrated system of emission reduction on the IC engine.

concentration emission limit.

The first alternative is to apply the 40 percent reduction to the highest uncontrolled $NO_x$ emission level within a fuel category. For example, Figure 4-49(c), which summarizes $NO_x$ emission reductions achieved by gas engines, lists 29 g/hp-hr as the highest uncontrolled $NO_x$ emission level. The application of a 40 percent reduction would lead to an emission level of about 17 g/hp-hr. As illustrated in Figure 9-16, if this level were selected as a standard of performance, 99 percent of production gas engines could easily meet the emission limit by reducing emissions by 40 percent or less. However, 69 percent of production engines would not have to reduce $NO_x$ emissions at all. Only one percent of production engines would have to reduce $NO_x$ emissions by more than 40 percent.

The second alternative is to apply the 40 percent reduction to the sales-weighted average uncontrolled $NO_x$ emission level. For example, the sales-weighted average uncontrolled $NO_x$ level for gas engines is 15 g/hp-hr. The application of a 40 percent reduction would lead to an $NO_x$ emission level of 9 g/hp-hr. As illustrated in Figure 9-16, if this level were selected as a standard of performance, 50 percent of production gas engines could meet the standard with 40 percent or less reduction in $NO_x$ emissions. However, 50 percent of production gas engines would be required to reduce $NO_x$ emission by greater than 40 percent. Only seven percent of production gas engines would not have to reduce $NO_x$ emissions at all.

The third alternative is to base the standard on a 40 percent reduction in $NO_x$ emissions from the sales-weighted average uncontrolled $NO_x$ emission level plus one standard deviation. For example, the sales-weighted average uncontrolled $NO_x$ emission level for gas production gas engines is 15 g/hp-hr and the standard deviation of the production gas engine data base (Appendix C) is 4 g/hp-hr. Thus, the application of a 40 percent reduction to the sum of these two values would lead to an emission level of 11 g/hp-hr. As illustrated in Figure 9-16, if this level were selected as a standard of performance, 84 percent of the production gas engines would not have to reduce $NO_x$ emission at all. Only 16 percent of the production gas engines would have to reduce $NO_x$ emissions by more than 40 percent.

Similarly, applying the three alternatives to dual-fuel engines yields results similar to those for gas engines. The highest uncontrolled $NO_x$ emission level from a dual fuel engine is 15 g/hp-hr. The application of a 40 percent reduction would lead to an emission level of 9 g/hp-hr. If this level were selected as a standard of performance, 98 percent of production dual-fuel engines could easily meet the emission limit by reducing $NO_x$ emissions by 40 percent or less. However, 62 percent would have to achieve no reduction in $NO_x$ emissions. Only two percent of production engines would have to reduce $NO_x$ emissions by more than 40 percent.

The sales-weighted average uncontrolled $NO_x$ emission level for dual-fuel engines is 8 g/hp-hr. The application of a 40 percent reduction would lead to a $NO_x$ emission level of about 5 g/hp-hr. If this level were selected as a standard of performance, 54 percent of the production dual-fuel engines could meet the standard by reducing $NO_x$ emissions 40 percent or less. Only 18 percent of the production dual-fuel engines would not have to reduce $NO_x$ emissions at all. Also, 46 percent of the production dual-fuel engines would be required to reduce $NO_x$ emissions by greater than 40 percent.

The standard deviation of the production dual-fuel engine data base is 3.2 g/hp-hr. Thus, the application of a 40 percent reduction to the sum of the sales-weighted average uncontrolled $NO_x$ emission level (8 g/hp-hr) and the standard deviation (3.2 g/hp-hr) would lead to an emission level of 7 g/hp-hr. If this level were selected as a standard of performance, 87 percent of the production dual-fuel engines could easily meet the emission limit by reducing emissions by 40 percent or less. However, 48 percent of the production gas engines would not have to reduce $NO_x$ emission at all. Only 13 percent of the production gas engines would have to reduce $NO_x$ emissions by more than 40 percent.

Finally, the application of the three alternatives to diesel engines also yields very similar results. The highest uncontrolled $NO_x$ emission level from a diesel engine is 19 g/hp-hr. The application of a 40 percent reduction would lead to in an emission level of about 11 g/hp-hr. If this level were selected as a standard of performance, 98 percent of production diesel engines could easily meet the emission limit by reducing emissions by 40 percent or less. However, 40 percent would have to achieve no reduction in $NO_x$ emissions. Two percent of production engines would be required to reduce $NO_x$ emission by more than 40 percent.

The sales-weighted average uncontrolled $NO_x$ emission level for diesel engines is 11 g/hp-hr. The application of a 40 percent reduction would lead to a $NO_x$ emission level of about 7 g/hp-hr. If this level were selected as a standard of performance, 56 percent of the production diesel engines could meet the standard by reducing $NO_x$ emissions 40

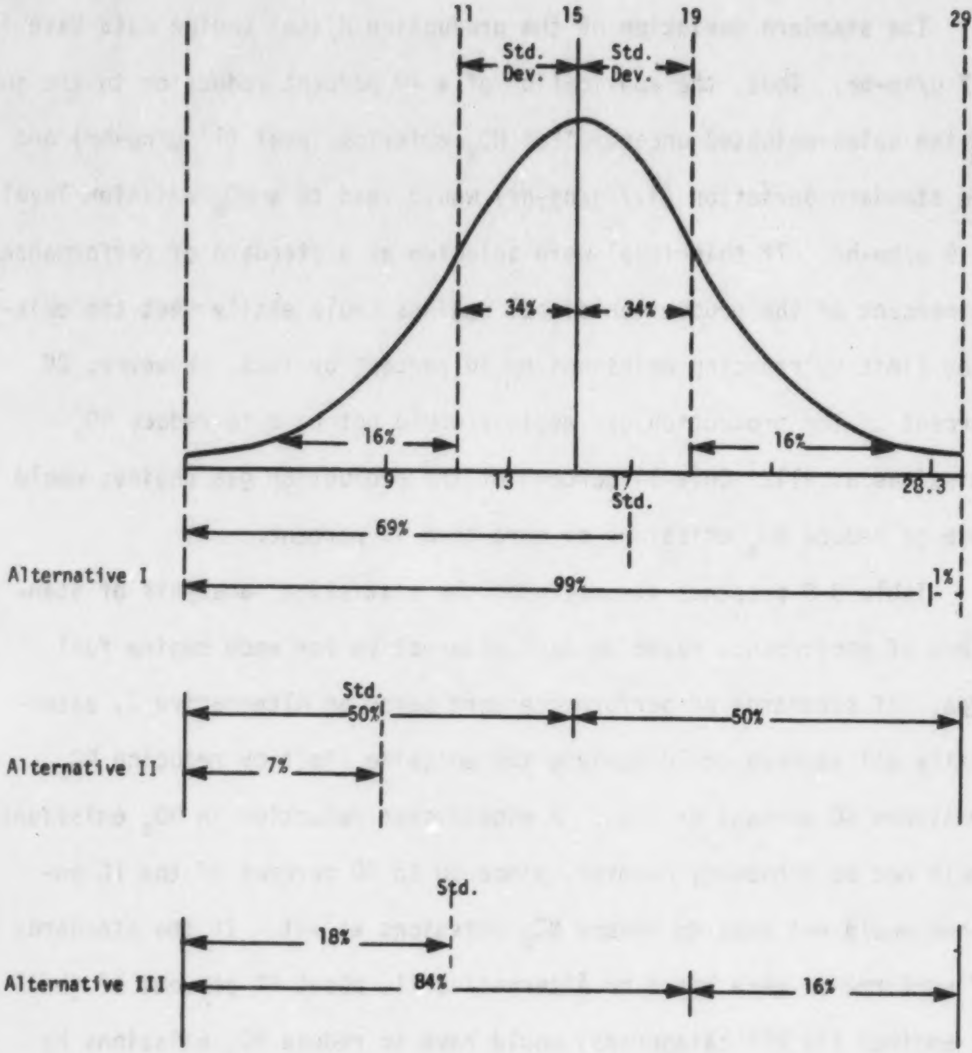

Figure 9-16. Statistical effects of alternative emission limits on gas engines.

percent or less. Only 14 percent of the production diesel engines would not have to reduce $NO_x$ emissions at all. However, 44 percent of the production diesel engines would be required to reduce $NO_x$ emission by greater than 40 percent.

The standard deviation of the production diesel engine data base is 3.7 g/hp-hr. Thus, the application of a 40 percent reduction to the sum of the sales-weighted uncontrolled $NO_x$ emission level (11 g/hp-hr) and the standard deviation (3.7 g/hp-hr) would lead to a $NO_x$ emission level of 9 g/hp-hr. If this level were selected as a standard of performance, 86 percent of the production diesel engines could easily meet the emission limit by reducing emissions by 40 percent or less. However, 29 percent of the production gas engines would not have to reduce $NO_x$ emissions at all. Only 14 percent of the production gas engines would have to reduce $NO_x$ emissions by more than 40 percent.

Table 9-9 presents a summary of the statistical analysis of standards of performance based on each alternative for each engine fuel type. If standards of performance were based on Alternative I, essentially all engines could achieve the emission limit by reducing $NO_x$ emissions 40 percent or less. A significant reduction in $NO_x$ emissions would not be achieved, however, since 50 to 70 percent of the IC engines would not have to reduce $NO_x$ emissions at all. If the standards of performance were based on Alternative II, about 50 percent of the IC engines (in all categories) would have to reduce $NO_x$ emissions by greater than 40 percent. Less than 10 percent would not have to reduce $NO_x$ emissions at all. Thus, this alternative would achieve a significant reduction in $NO_x$ emissions from new sources. If standards of

TABLE 9-9. SUMMARY OF STATISTICAL ANALYSES OF ALTERNATE EMISSION LIMITS

GAS ENGINES

| Alternative | I | II | III |
|---|---|---|---|
| Standard | 17 | 9 | 11 |
| Percent required to apply less than or equal to 40 percent control | 99 | 50 | 84 |
| Percent required to do nothing | 69 | 7 | 18 |
| Percent required to apply more than 40 percent control | 1 | 50 | 16 |

DUAL-FUEL ENGINES

| Alternative | I | II | III |
|---|---|---|---|
| Standard | 9 | 5 | 7 |
| Percent required to apply less than or equal to 40 percent control | 98 | 54 | 87 |
| Percent required to do nothing | 62 | 18 | 48 |
| Percent required to apply more than 40 percent control | 2 | 46 | 13 |

DIESEL ENGINES

| Alternative | I | II | III |
|---|---|---|---|
| Standard | 11 | 7 | 9 |
| Percent required to apply less than or equal to 40 percent control | 98 | 56 | 86 |
| Percent required to do nothing | 50 | 4 | 29 |
| Percent required to apply more than 40 percent control | 2 | 44 | 14 |

performance were based on Alternative III, the results would be similar to those achieved with Alternative I. About 85 percent of engines could easily meet the standards by reducing $NO_x$ emissions by less than 40 percent. About 20 to 30 percent of IC engines would not have to reduce $NO_x$ emissions at all and about 15 percent of IC engines would have to reduce $NO_x$ emissions by more than 40 percent.

In light of the high priority discussed earlier which has been directed toward reducing $NO_x$ emissions and the significance of IC engines in terms of their contribution to $NO_x$ emissions from stationary sources, the second alternative was chosen for selecting the $NO_x$ emission concentration limit. This approach will achieve the greatest reduction in $NO_x$ emissions from new IC engines.

## Selection of Limits

A concentration (ppm) format was selected for the standards. Consequently, the brake-specific $NO_x$ emission limits corresponding to the second alternative for selecting numerical emission limits (i.e., gas - 9 g/hp-hr; dual-fuel - 5 g/hp-hr; diesel - 7 g/hp-hr) must be converted to concentration limits (corrected to 15 percent oxygen). This may be done by dividing the brake-specific volume of $NO_x$ emissions by the brake specific total exhaust gas volume. Determining the brake-specific volume of $NO_x$ emissions is straightforward. Determining the brake-specific total exhaust gas volume is more complex, in that the brake-specific exhaust flow and the exhaust gas molecular weight are unknown. Knowing the fuel heating value and composition, the brake-specific fuel consumption, and assuming 15 percent excess air, however, defines these unknowns. (The complete derivation is explained in detail in Appendix C-5).

Combining these factors leads to the following conversion factor:

$$NO_{x_{15}} = \frac{\left(\frac{10^6}{46}\right) \times BSNO_x}{\left(\frac{16.6 + 3.29\,Z}{12.0 + Z}\right) \times BSFC}$$

where:

$NO_{x_{15}}$ = $NO_x$ concentration ppmv corrected to 15-percent oxygen on a dry basis.

$BSNO_x$ = Brake-specific $NO_x$ emission, g/hp-hr.

BSFC = Brake-specific fuel consumption, g/hp-hr.

Z = Hydrogen/Carbon ratio of the fuel.

The numerator is the brake specific volume of $NO_x$ emissions multiplied by $10^6$ in order to convert the decimal equivalent to ppm. In the denominator, the brake-specific total dry exhaust gas volume with 15 percent excess oxygen is expressed as a function of the fuel's hydrogen/carbon ratio and the brake-specific fuel consumption. The fuel consumption has been converted from Btu/hp-hr to g/hp-hr using the fuel's lower heating value (LHV).

For natural gas, a hydrogen to carbon (H/C) ratio of 3.5 and an LHV of 20,000 Btu/lb was assumed. Diesel ASTM-2 has a H/C ratio of 1.8 and an LHV of 18,320 Btu/lb.

Using the above equation, plots of $BSNO_x$ versus ppm were generated for each fuel type. The uncontrolled values of $BSNO_x$ and BSFC were used for each engine, producing plots similar to Figure 9-17. As shown, agreement between this equation and actual emission data relating

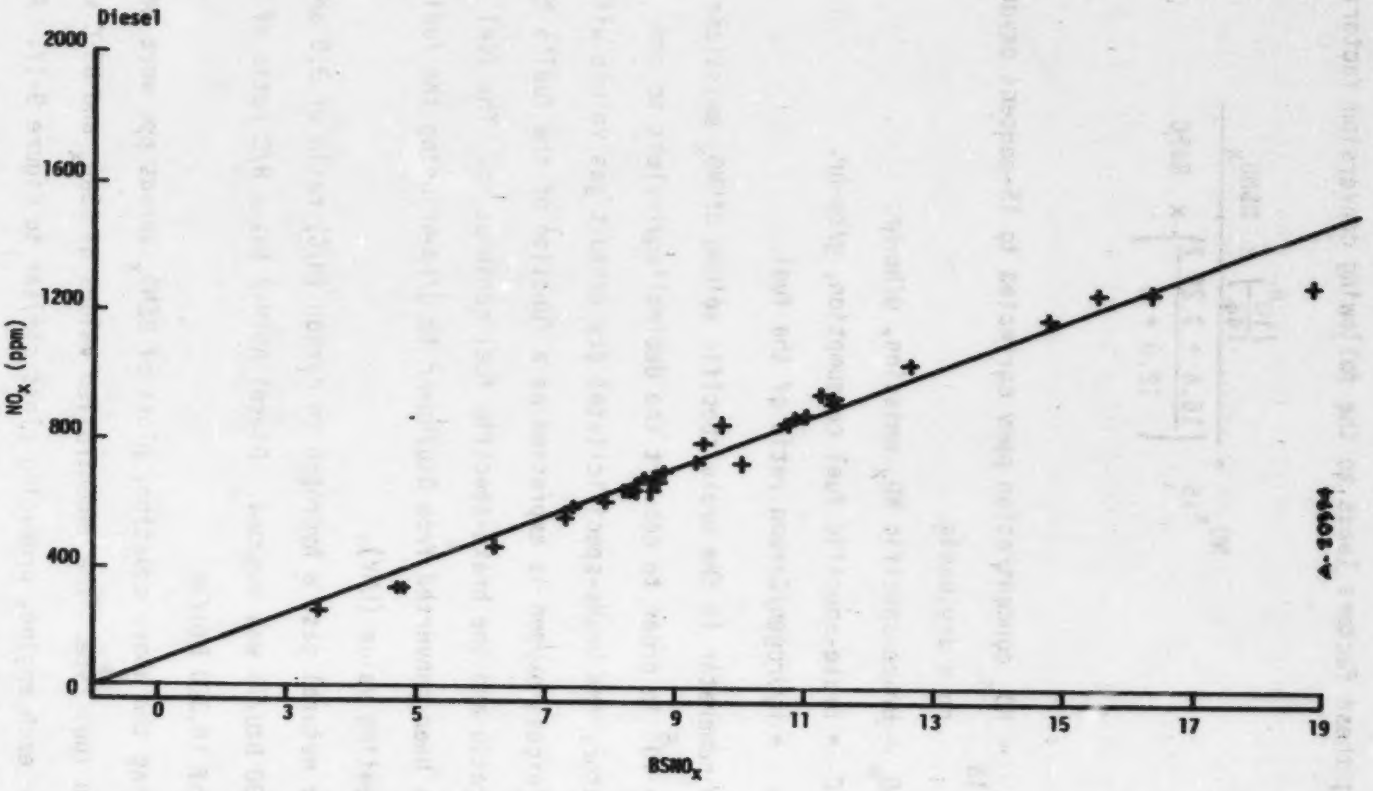

Figure 9-17. BSNO$_x$ converted to NO$_{x_{15}}$ and best fit straight line determined.

$NO_x$ emissions concentrations to $BSNO_x$ is excellent. Comparison of this conversion factor with available raw data also shows excellent agreement (see Appendix C.5). Applying this conversion factor to the brake-specific emission limits associated with the second alternative for $NO_x$ emission limits leads to the $NO_x$ concentration emission limits for large stationary internal combustion engines summarized in Table 9-10.

These emission limits have been rounded upward to the nearest 100 ppm to include a "margin" to allow for source variability. The standard for diesel engines has also been applied to dual-fuel engines. If a separate emission limit had been selected for dual-fuel engines, the corresponding numerical $NO_x$ concentration emission limit would be 400 ppm. Sales of dual-fuel engines have ranged from 17 to 95 units annually over the past five years, with a general trend of decreasing sales. Dual-fuel engines serve the same applications as diesel engines, and new dual-fuel engines will likely operate primarily as diesel engines due to increasingly limited natural gas supplies. Thus, combining of dual-fuel engines with diesel engines for standards of performance will have little adverse impacts and will simplify enforcement of the standards of performance.

TABLE 9-10. NUMERICAL $NO_x$ CONCENTRATION EMISSION LIMITS FOR LARGE STATIONARY INTERNAL COMBUSTION ENGINES

| Engine | $NO_x$ Emission Limit |
|---|---|
| Gas | 700 ppm |
| Diesel/dual-fuel | 600 ppm |

As discussed in Section 4.2.1, the effect of ambient atmospheric conditions on $NO_x$ emission from large stationary internal combustion engines can be significant. Therefore, to enforce the standards uniformly, $NO_x$ emissions must be determined relative to a reference set of ambient conditions. All existing ambient correction factors were reviewed that could potentially be applied to large stationary internal combustion engines to correct $NO_x$ emissions to standard conditions. A detailed discussion of this review is presented in Section 4.2 and Appendix C.2.

The correction factors that were selected for both spark ignition (SI) and compression ignition (CI) engines are presented in Table 4-2, which is reproduced below as Table 9-11. For the compression ignition engines (i.e., diesel and dual-fuel), a single correction factor for both temperature and humidity was selected. Constants for use in this correction factor, which have been experimentally derived, are summarized in the table. For spark ignition engines (i.e., gas), separate correction factors were selected for humidity and temperature, and measured $NO_x$ emissions are corrected to reference ambient conditions by multiplying these two factors together. No correction factor was selected for changes in ambient pressure, because no generalized relationship could be determined for the very limited data that were available. These correction factors represent the general effects of ambient temperature and relative humidity on $NO_x$ emissions, and will be used to adjust measured $NO_x$ emissions during any performance test to determine compliance with the numerical emission limit.

TABLE 9-11 EXISTING INTERNAL COMBUSTION ENGINE AMBIENT CORRECTION FACTORS FOR APPLICATION TO LARGE-BORE ENGINES

| Fuel | Correction Factor |
|---|---|
| Diesel and Dual-Fuel | $K = 1/(1 + 0.00235(H - 75) + 0.00220\ (T - 85))$ |
| Gas | $K = (K_H)\ (K_T)$<br>$K_H = 0.844 + 0.151\ (\frac{H}{100}) + 0.075\ (\frac{H}{100})^2$<br>$K_T = 1 - (T - 85)(0.0135)$ |

where:

$H$ = observed humidity, grains $H_2O$/lb dry air

$T$ = observed inlet air temperature, °F

Since the recommended factors may not be applicable to certain engine models, as an alternative to the use of these correction factors, engine manufacturers, owners, or operators may elect to develop their own ambient correction factors. All such correction factors, however, must be substantiated with data and then approved for use by EPA in determining compliance with the $NO_x$ emission limits. The ambient correction factor will be applied to all performance tests, not only those in which the use of such factors would reduce measured emission levels.

As discussed in the Standards Support and Environmental Impact Statement, "Proposed Standards of Performance for Stationary Gas Turbines," EPA-450/2-77-017a, the contribution to $NO_x$ emissions by the conversion of fuel-bound nitrogen in heavy fuel to $NO_x$ can be significant for stationary gas turbines. The organic $NO_x$ contribution to total gas turbine $NO_x$ emission is complicated by the fact that the percentage of fuel-bound nitrogen converted to $NO_x$ emissions varies with the fuel-bound nitrogen level. Figure 9-18 illustrates the variation in conversion of fuel-bound nitrogen to $NO_x$ emissions with the fuel-bound nitrogen level of the fuel. While this figure is based on very limited data, it indicates that the percentage of fuel-bound nitrogen converted to $NO_x$ emission decreases as the fuel-bound nitrogen level increases. Below a fuel-bound nitrogen level of about 0.05 percent, essentially 100 percent of the fuel-bound nitrogen is converted to $NO_x$. Above a fuel-bound nitrogen level of about 0.4 percent, only about 40 percent is converted to $NO_x$. Using Figure 9-18, an estimate of the effect on controlled $NO_x$ emission levels of firing fuels with various fuel-bound nitrogen levels can be made.

As discussed in the Standards Support and Environmental Impact Statement, "Proposed Standards of Performance for Stationary Gas Turbines," EPA-450/2-77-017a, assuming a fuel with 0.25 weight percent fuel-bound nitrogen (which allows approximately 50 percent availability of domestic heavy fuel oil), controlled $NO_x$ emission would increase by about 50 ppm due to the contribution to $NO_x$ emissions of fuel-bound nitrogen. In gas turbines, this contribution was significant when compared to the proposed emission limit of 75 ppm. It can be assumed that the

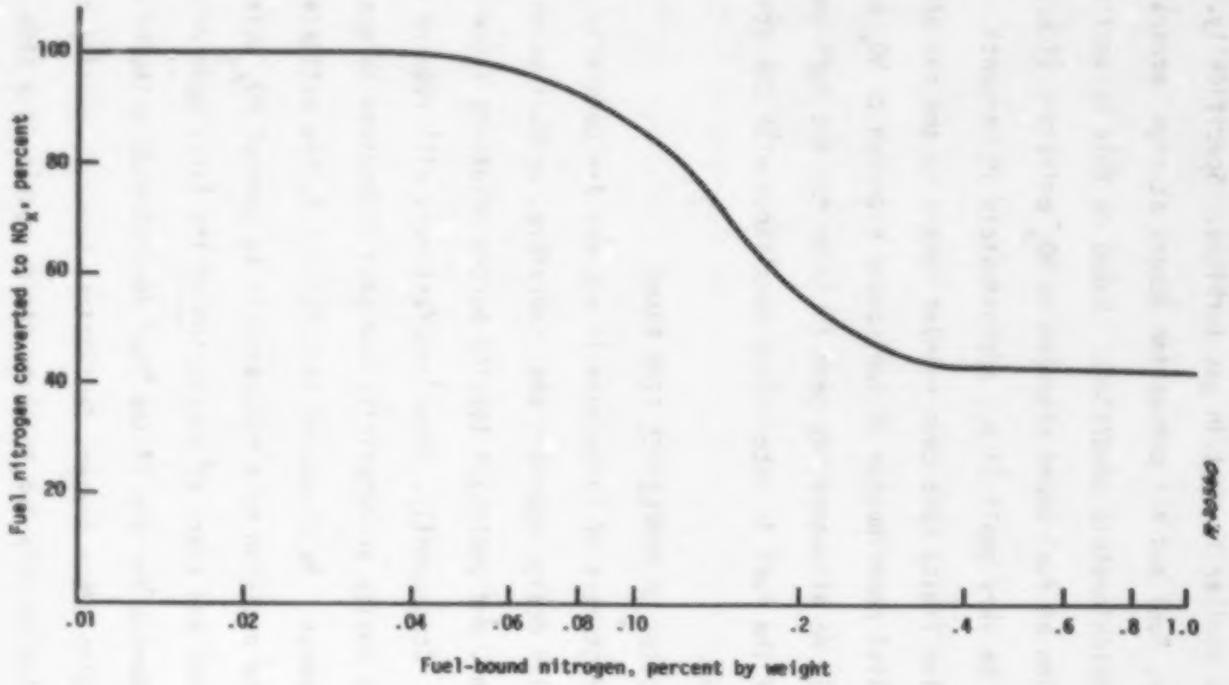

Figure 9-18. Variation in conversion of fuel-bound nitrogen to organic NO$_x$ in a gas turbine.

conversion of fuel-bound nitrogen to $NO_x$ in large internal combustion engines is similar to that in gas turbines. Specifically, in both facilities, fuel and air combustion occurs at high temperatures and at similar stoichiometric conditions. Based on this assumption, the contribution of fuel-bound nitrogen to $NO_x$ emissions (i.e., 50 ppm), is likely to be very small (i.e., approximately 10 percent). Given that the emission limits have been rounded upward to the nearest 100 ppm and the potential contribution of fuel-bound nitrogen to $NO_x$ emissions is very small, no allowance has been included for the fuel-bound nitrogen content of the fuel in determining compliance with the standards of performance.

## 9.7 SELECTION OF COMPLIANCE TIME FRAME

Manufacturers of large-bore IC engines are generally committed to a particular design approach and, therefore, conduct extensive research, development, and prototype testing before releasing a new engine model for sale. Consequently, these manufacturers will require some period of time to modify or reoptimize and test IC engines to meet standards of performance. As discussed in Section 6.3, the estimated time span between the decision by a manufacturer to control $NO_x$ emissions from an engine model and start of production of the first controlled engine is about 15 months for any of the four demonstrated emission control techniques. With their present facilities, however, testing can typically be conducted on only two to three engine models at a time. Since most manufacturers produce a number of engine models, additional time is required before standards of performance become effective. In addition, a number of manufacturers produce their most popular engine models at a fairly steady rate of production and satisfy fluctuating demand from inventory. Consequently, additional time is necessary to allow manufacturers to sell their current inventory of uncontrolled IC engines before they must comply with standards of performance.

It is estimated that about 30 months delay in the effective date of the standard is appropriate to allow manufacturers time to comply with the proposed standards. In addition, in light of the stringency of the standards (i.e., many models will have to reduce $NO_x$ emissions by more than 40 percent), this time period provides the flexibility of manufacturers to develop and use combinations of the control techniques upon which the standards are based or other control techniques. Consequently, 30 months from the date of proposal is selected as the delay period for implementation of these standards on large stationary internal combustion engines.

## 9.8 MODIFICATION/RECONSTRUCTION

A discussion of the modification and reconstruction regulations and how they pertain to the internal combustion engine industry can be found in Chapter 5. Since few modified or reconstructed internal combustion engines are anticipated, the modification and reconstruction regulations will have little impact. The demonstrated $NO_x$ reduction techniques, however, are as effective in reducing emissions of $NO_x$ from modified or reconstructed internal combustion as from new internal combustion engines. Thus, modified or reconstructed internal combustion engines merit no special allowance in the standards of performance.

## 9.9 SELECTION OF PERFORMANCE TEST METHOD

A performance test method is required to determine whether an engine complies with the standards of performance. Reference Method

20, "Determination of Nitrogen Oxides, Sulfur Dioxide, and Oxygen emissions from Stationary Gas Turbines," is proposed as the performance test method for IC engines. Reference Method 20 has been shown to provide valid results. Consequently, it was decided that rather than a totally new reference test method being developed, Reference Method 20 would be modified for use on IC engines.

The changes and additions to Reference Method 20 required to make it applicable for testing of internal combustion engines include (by section):

1. Principle and Applicability. Sulfur dioxide measurements are not applicable for internal combustion engine testing.

6.1 Selection of a sampling site and the minimum number of traverse points.

6.11 Select a sampling site located at least five stack diameters downstream of any turbocharger exhaust, crossover junction, or recirculation take-offs and upstream of any dilution air inlet. Locate the sample site one meter or three stack diameters (whichever is less) upstream of the gas discharge to the atmosphere.

6.1.3 A preliminary $O_2$ traverse is not necessary.

6.2 Cross-sectional layout and location of traverse points use a minimum of 3 sample points located at positions of 16.7, 50 and 83.3 percent of the stack diameter.

6.3.1.4 Record the data required on the engine operation record on Figure 20.6 of Reference Method 20. In addition, record (a) the intake manifold pressure; (b) the intake manifold temperature; (c) rack position; (d) engine speed; and (e) injector or spark fuming. (The water or steam injection rate is not applicable to internal combustion engines.)

$NO_x$ emissions measured by Reference Method 20 will be affected by ambient atmospheric conditions. Consequently, measured $NO_x$ emissions would be adjusted during any performance test by the ambient condition correction factors discussed earlier, or by custom correction factors approved for use by EPA.

The performance test may be performed either by the manufacturer or at the actual user operating site. If the test is performed at the manufacturer's facility, compliance with that performance test will be sufficient proof of compliance by the user as long as the engine operating parameters are not varied during user operation from the settings under which testing was done.

## 9.10 SELECTION OF MONITORING REQUIREMENTS

To provide a means for enforcement personnel to ensure that an emission control system installed to comply with standards of performance is properly operated and maintained, monitoring requirements are generally included in standards of performance. For stationary internal combustion engines, the most straightforward means of ensuring proper operation and maintenance would be to monitor $NO_x$ emissions released to the atmosphere.

Installed costs, however, for continuous $NO_x$ monitors are approximately \$25,000.$^{(32)}$ Thus, the cost of continuous $NO_x$ emission monitoring is considered unreasonable for IC engines since most large stationary IC engines only cost from \$50,000 to \$3,000,000 (i.e., 1000 hp gas production engine and 20,000 hp electrical generation engine).

A more simple and less costly method of monitoring is measuring various engine operating parameters related to $NO_x$ emissions. Consequently, monitoring of exhaust gas temperature was considered since this parameter could be measured just after the combustion process where $NO_x$ is formed. However, a thorough investigation of this approach showed no simple correlation between $NO_x$ emissions and exhaust gas temperature.

A qualitative estimate of $NO_x$ emissions, however, can be developed by measuring several engine operating parameters simultaneously, such as spark ignition or fuel injector timing, engine speed, and a number of other parameters.(33) These parameters are typically measured at most installations and thus should not impose an additional cost impact. For these reasons, the emission monitoring requirements included in the proposed standards of performamce require monitoring various engine operating parameters.

For diesel and dual-fuel engines, the engine parameters to be monitored are: (1) intake manifold temperature; (2) intake manifold pressure; (3) rack position; (4) fuel injector timing; and (5) engine speed. Gas engines would require monitoring of: (1) intake manifold temperature; (2) intake manifold pressure; (3) fuel header pressure; (4) spark timing; and (5) engine speed.

Another parameter that could be monitored for gas engines is the fuel heat value, since it can affect $NO_x$ emissions significantly. Because of the high costs of a fuel heating value monitor, and the fact that many facilities can obtain the lower heating value directly from the gas supplier, monitoring of this parameter would not be required.

The operating ranges for each parameter over which the engine could operate and in which the engine could comply with the emission limit would be determined during the performance test. Once established, these parameters would be monitored to ensure proper operation and maintenance of the emission control techniques employed to comply with the standards of performance.

For facilities having an operator present everyday, these operating parameters would be recorded daily. For remote facilities where an operator is not present every day, these parameters would be recorded weekly. The owner/operator would record the parameters, and if these parameters include values outside the operating ranges determined during the performance test, a report would be submitted to the Administrator on a quarterly basis identifying these periods as excess emissions. Each excess emission report would include the operating ranges for each parameter as determined during the performance test, the monitored values for each parameter, and the ambient air conditions.

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# APPENDIX A

# EVOLUTION OF PROPOSED STANDARDS

This study to develop proposed standards of performance for new stationary reciprocating internal combustion engines began on June 24, 1974. In March 1976, draft Standards Support Document and Environmental Impact Statement (SSEIS) was presented to the National Air Pollution Control Techniques Advisory Committee (NAPCTAC). This draft had been prepared by the Aerotherm Division of Acurex Corporation, Mountain View, California under Contract 68-02-1318, Task No. 7. Randy Seiffert, Standards Support Criteria Pollutants Section, Industrial Studies Branch, Emission Standards and Engineering Division (ESED), Office of Air Quality Planning and Standards (OAQPS), U.S. EPA, was the Project Engineer. A revised draft SSEIS was prepared by Acurex/Aerotherm under Contract 68-02-2530 beginning in October 1976, under the direction of Mr. Douglas Bell (Project Engineer) and Mr. Fred Porter (Section Chief), Standards Development Branch, Regulations Preparation Section Branch, ESED/OAQPS, U.S. EPA. Preliminary work for this revision was accomplished under Contract 68-01-3158, Task 14, beginning in July 1976, under the direction of Mr. John McDermon, Standards Development Branch, ESED/OAQPS, U.S. EPA.

The first step in this investigation was to initiate a two-part literature survey, one part being directed toward technical questions and the other toward a characterization of the industry. The technically-oriented survey sought information on the best control technologies available for stationary engines. The bibliographical services of APTIC, the Engineering Index, and the indices of the Transactions of the Society of Automotive Engineers (SAE) and of the American Society of Mechanical Engineers (ASME) were studied, and appropriate papers obtained. As a result of this survey, contacts were initiated with staff members at Southwest Research Institute (SwRI), who had conducted a study for the American Gas Association (AGA) on natural gas fueled engines used in pipeline compressor stations. SwRI has also investigated emissions from a variety of engines used in mobile applications, ranging from small gasoline units to large diesel locomotive engines. The Bartlesville Energy Research Center of the Bureau of Mines, U.S. Department of Interior, was found to have conducted a significant amount of R&D work on emissions reductions from engines, mainly 100- to 500-hp diesel and gasoline units. In addition to these studies, most published research was concerned with emissions from automobile engines.

Business indices, such as Moody's and Dunn and Bradstreet, were consulted to identify the domestic manufacturers of engines for stationary applications. However, the trade journals were found to be much more helpful for this purpose.[1/] These included the Oil and Gas Journal, Automotive Engineering, and Power. In addition, the Diesel and Gas Turbine Worldwide Catalog was especially useful(1). Other sources were Federal Power Commission (FPC) publications, which contained information about engine-driven electric generators, various AGA documents, which gave information on the gas pipeline

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[1/] However, these indices, plus statistical data published by Fortune Magazine, were used to evaluate the financial position of the engine manufacturers.

industry's use of engines, and statistical surveys by the U.S. Department of Agriculture, which provided some data on agricultural pumping requirements. Some annual production statistics were obtained from the Commerce Department's series of Current Industrial Reports, but frequently these documents did not divide production figures into sufficient subgroups to correlate a production trend with a particular application. Therefore, the majority of the production, marketing, and application data were obtained by direct telephone or written contacts with marketing representatives, application engineers, or environmental managers within companies who sell engines for stationary application.

Direct contacts with engine manufacturers also provided much of the data on emissions from uncontrolled engines, control technologies, and emissions from controlled engines. In addition, a report by McGowin(2) provided useful information about large-bore engine applications and utilization rates and one by Roessler, et al.,(3) presented a thorough discussion of control technologies used in mobile applications. Additional information regarding stationary applications, control technologies, and their costs were received from manufacturers in response to official requests for data by the Director, Emission Standards and Engineering Division, U.S. EPA. These requests were sent to manufacturers under the authority specified in Section 114 of the Clean Air Act. Most of the data on the effectiveness of the various controls come from laboratory experiments at the manufacturer's plants or, in a few cases, at special test centers such as at SwRI or Bartlesville.

An extensive telephone survey was conducted during July and August 1974 among local and state air pollution control authorities (see Table A-1) in an attempt to locate potential examples of "best demonstrated control

Table A-1. AIR POLLUTION CONTROL AGENCIES CONTACTED IN SEARCH OF CONTROLLED STATIONARY ENGINES

| State | | Office Contacted[a] |
|---|---|---|
| Arizona | 1. | Department of Health, Division of Air Pollution Control |
| California | 1. | Bay Area APCD |
| | 2. | Kern County APCD |
| | 3. | Los Angeles APCD |
| | 4. | San Bernardino APCD |
| | 5. | San Diego APCD |
| Colorado | 1. | Department of Health |
| | 2. | City and County of Denver Health Department |
| | 3. | Tri-County Health Department |
| Florida | 1. | Department of Pollution Control: Air, Noise, and Solid Waste |
| | 2. | Dade County |
| Illinois | 1. | Illinois EPA — Air Pollution (Chicago) |
| Missouri | 1. | St. Louis APCD |
| New Jersey | 1. | Department of Environmental Protection (Trenton Central Office) |
| New York | 1. | Department of Environmental Conservation |
| | 2. | New York City Department of Air Resources |
| Oklahoma | 1. | Oklahoma City — County Health Department |
| | 2. | Tulsa City — County Health Department |
| Oregon | 1. | Department of Environmental Quality |
| | 2. | Lane County Air Pollution Authority |
| | 3. | Mid-Willamette Valley Air Pollution Authority |
| Pennsylvania | 1. | Alleghany County Air Pollution Control |
| Texas | 1. | Air Control Board — Austin |
| | 2. | Air Control Board — Houston |

[a] APCD — Air Pollution Control District

technology." Based on this survey, which was planned to include all the geographical areas known to have relatively stringent regulations, it was concluded that no installed engines were currrently subjected to any emission limitations other than for visible emissions. The only exception was the 140 lb/hr rule for $NO_x$ from new engines in Southern California (see Section 2.2). Since most uncontrolled engines under 4000 hp can meet this regulation, and since no larger engines are known to have been installed in that area recently, that rule did not lead us to any examples of field-installed controlled engines. Therefore, the evaluation of control technology was to be based upon published emission data and data obtained from engine manufacturers.

Since no installed engines were found which could be cited as examples of units with "best" controls, no emission tests were conducted. However, a San Jose, California sewage plant (San Jose Water Pollution Control Plant) was visited to give a better understanding of the operational flexibilities available to users of large-bore engines.(4) This plant had six gas engines and six dual-fuel units.

The direct contacts were supplemented by visits to the manufacturers listed in Table A-2. The purpose of these visits was to obtain more information concerning the status of their R&D efforts in emission reductions, their experience with the commonly proposed control technologies, their estimates of the cost and time required to incorporate such controls on their engines, and the importance of the stationary market place in their total sales.

The group meetings that were held with representatives of engine manufacturer's associations are listed in Table A-3. In general, the meetings served to inform these representatives of the purpose of performance standards for new sources, the process of developing a standard, including

Table A-2. ENGINE MANUFACTURERS VISITED DURING PREPARATION OF THE STANDARDS SUPPORT DOCUMENT

| Company | Location | Date Visited | Category | Reason for Selection |
|---|---|---|---|---|
| Caterpillar Tractor Co.[a] | Peoria, IL | 8/27/74 | Medium Bore | Make precombustion chamber engines; have published several technical papers on emissions |
| Cummins Engine Co.[a] | Columbus, IN | 8/28/74 | Medium Bore | One of best known manufacturers of truck engines; have published several technical papers on emissions |
| Enterprise Engine Div. | Oakland, CA | 7/18/74 | Large Bore | One of largest manufacturers in large-bore category; close to Aerotherm/Acurex |
| Waukesha Motor Co. | Waukesha, WI | 7/25/74 | Medium and Large Bore | Market widest range of engine sizes |
| Worthington | Buffalo, NY | 9/1/74 | Large Bore | Manufacturer of large-bore gas engines |

[a] Aerotherm staff engineer accompanied by EPA Project Officer

## TABLE A-3. MEETINGS WITH ENGINE MANUFACTURERS ASSOCIATIONS

| Date | Association | Meeting Location | EPA/Aerotherm Attendance | Report Date |
|---|---|---|---|---|
| 8/28/74 | Diesel Engine Manufacturers Assoc. (DEMA) | EPA (Durham, NC) | EPA | 9/3/74 |
| 11/21/74 | Engine Manufacturers Assoc. (EMA) | EPA (Durham, NC) | EPA and Aerotherm[a] | 11/26/74 |
| 4/17-18/75 | Engine Manufacturers Assoc. (EMA) | Aerotherm/Acurex (Mountain View, CA) | Aerotherm | 4/22/75 (2 reports) |
| 6/10/76 | American Gas Assoc.[b] | EPA (Durham, NC) | EPA | 6/26/75 |
| 5/28/76 | American Gas Assoc. | EPA (Durham, NC) | EPA | 4/30/76 |
| 5/4/76 | Colt Industries | EPA (Durham, NC) | EPA | 5/18/76 |
| 5/12/76 | Engine Manufacturers Assoc. (EMA) | EPA (Durham, NC) | EPA and Aerotherm | 5/18/76 |
| 5/13/76 | Diesel Engine Manufacturers Assoc. (DEMA) | EPA (Durham, NC) | EPA and Aerotherm | 5/18/76 |
| 5/14/76 | General Motors Corporation | EPA (Durham, NC) | EPA and Aerotherm | 5/18/76 |

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[a] Meeting report prepared by Aerotherm

[b] Supervisory committee for Project PR-15-61, "$NO_x$ Research". Accompanied by staff from Southwest Research Institute

informal and formal opportunities for input from all interested parties, and the reasons for considering that stationary engines should be regulated by performance standards. The meetings were also used to request information of a general nature, such as data on the effects of variable ambient conditions (temperature, humidity, etc.) on emissions, appropriate test cycles (based on their knowledge of engine usage patterns) possible engine operating parameters that are known to be directly and uniquely related to emission levels and could be used in lieu of emission testing for compliance monitoring. As is also shown in Table A-3, representatives of three industry organizations met with EPA and commented on the draft SSEIS after it was presented at the March 1976 NAPCTAC meeting in Washington, D.C.

Several meetings were held between Acurex/Aerotherm and EPA staff during the development of the draft document. In the first half of 1978, two public meetings were held to discuss the proposed standards, one before the NAPCTAC committee and the other with the engine manufacturers. These are listed in Table A-4 along with a brief description of the purpose, conclusion, and any redirections.

In early 1978, Acurex was assigned the responsibility for writing the rationale chapter of the SSEIS, the preamble to the standard, and the regulation itself. In addition, Acurex was also assigned responsibility for establishing monitoring and emission testing procedures. This work was performed with the advice and review of EPA/OAQPS.

Review of the original draft Standards Support Document and Environmental Impact Statement by the Industrial Studies Branch, Emission Standards and Engineering Division, OAQPS was completed on December 20, 1975, and the final draft was forwarded to the Industrial Studies Branch by Acurex/Aerotherm on March 17, 1976. The revised draft SSEIS was reviewed by the

Standards Development Branch, Emission Standards and Engineering Division, OAQPS by March 1978 and a final SSEIS was forwarded to the Standards Development Branch by Acurex/Aerotherm in July 1978.

## TABLE A-4. MAJOR PROJECT REVIEWS AND DECISION-RELATED COMMUNICATIONS

| Date | Format[a] | Purpose | Conclusions/Redirections |
|---|---|---|---|
| 7/23-24/74 | EPA-Aerotherm meeting; report prepared by EPA (August 5, 1974) | Review proposed Work Plan prepared by Aerotherm | Guidance provided to Aerotherm on procedural matters, including allowable and effective methods of collecting emission and marketing data, precautions to consider when requesting and evaluating user or manufacturer supplied emission data, air pollution control agencies that might regulate engines, protocol of interacting with users and manufacturers, desire to determine if one standard could be set rather than separate standards for each engine type and fuel, etc. Decision made to collect as much information as possible by telephone and individual follow-up letter rather than by questionnaire canvas of whole industry. |
| 9/19/74 | Memorandum (Industrial Studies Branch to Office of Enforcement and General Counsel) | Request legal opinion on time-phased standards, ability to treat standards on total hydrocarbons as standards on a criterion pollutant | See December 18, 1974 for response |
| 9/27/74<br>10/1/74 | Telephone conversations (EPA Project Officer and Aerotherm)[b] | Relay decision by EPA (Industrial Studies Branch) to solve problem of insufficient data by sending "Section 114" letters to selected engine manufacturers | Aerotherm to provide EPA with list of manufacturers who had not yet been contacted formally and were believed to have useful information (sent October 10, 1974 - October 24, 1974) |

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[a] All meetings held in EPA offices, Durham, NC unless otherwise noted.

[b] Many other telephone conversations occurred during which procedural details were discussed. These are not documented here.

TABLE A-4. Continued

| Date | Format[a] | Purpose | Conclusions/Redirections |
|---|---|---|---|
| 12/18/74 | Memorandum (Office of General Counsel to Industrial Studies Branch) | Response to ISB memos of September 19 and November 27, 1974 | Mandatory certification and recall program not permitted under Section 111, Section 114 binding on manufacturers for emission and cost to control data, effective date of standard need not be proposal date, and a standard on total hydrocarbons would probably not be a standard on a noncriterion pollutant |
| 2/10-11/75 | EPA-Aerotherm meeting; reports prepared by Aerotherm (Feb. 13, 1975) and EPA (Feb. 25, 1975) | To present status of project to ESED staff and discuss alternatives considered in major decision areas (selection of affected facilities, selection of pollutants for control, and proposed format for the standard) | Retabulate uncontrolled and controlled data by engine and fuel type. Consider need to propose separate standards by engine type and fuel based on economic impact. Examine technological and cost impact of subjecting standby engines to standards. Include some additional factors in rationale for proposed pollutants to be controlled and format of standard |
| 3/28/75 | Meeting with Director, Emission Standards and Engineering Division, OAQPS (ISB report dated April 3, 1975) | Report on status of project to Division, present recommendations to limit source category to engines over 500 hp, and identify anticipated problems | Continue to develop standards of performance for stationary engines over 500 hp. A possible result of the standard development study may be to recommend R&D program to Industrial Environmental Research Laboratory, RTP. |
| 4/4/75 | Letter (EPA Project Officer to Aerotherm) | Report on March 28 meeting with Director, ESED | Reorganize format of draft document to minimize discussion of engines less than 500 hp and limit the development of standards to engines over 500 hp. |
| 4/30/75 | Letter (EPA Project Officer to Aerotherm) | Transmit additional instructions on format of report | Incorporate minor changes as requested |
| 7/30/75 | Letter (EPA Project Officer to Aerotherm) | Indicate changes in drafts of reports submitted to Project Officer for review | Limit standard to engines greater than 350 CID/cyl |

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[a] All meetings held in EPA offices, Durham, NC unless otherwise noted.

[b] Many other telephone conversations occurred during which procedural details were discussed. These are not documented here.

## TABLE A-4. Continued

| Date | Format[a] | Purpose | Conclusions/Redirections |
|---|---|---|---|
| 3/17/76 | Meeting of National Air Pollution Control Techniques Advisory Committee (NAPCTAC) in Washington, D.C. | To present to NAPCTAC and industry the recommended standards of performance for stationary reciprocating IC engines and the background information leading to this recommendation | Industry to review the Standards Support and Environmental Impact Statement and meet with EPA in May 1976 |
| 5/13/76 | Meeting with the Diesel Engine Manufacturers Association (DEMA) | DEMA presented their comments and recommendations on emission standards for stationary IC engines | Since substantially more emission data were now available, "Section 114" letters were to be prepared to request this information from large-bore engine manufacturers |
| 6/16/76 | "Section 114 Request for Information" letter sent to large-bore (>350 CID/cyl) engine manufacturers | To obtain additional emissions data, market information, and control cost information | Acurex/Aerotherm to incorporate additional data in the SSEIS. Aerotherm received responses from "114" letters on 8/20/76 |
| 9/17/76 | EPA-Aerotherm meeting; report prepared by EPA (September 29, 1976) | To discuss applicability of new data (received from "114" responses) to the data base of the draft SSEIS, and to discuss revisions of the draft SSEIS | Aerotherm plan for incorporating additional data approved |
| 11/17/76 | EPA-Aerotherm meeting at Mountain View, CA | To discuss expansion and refinement of existing emissions data base | Aerotherm to document work concerning additional data, ambient correstions measurement procedures, and field applicability of control techniques in a supplemental report |
| 1/28/77 | Engine Manufacturers Association (EMA)-Aerotherm meeting at Mountain View, CA | To update EMA members on Aerotherm's revisions to draft SSEIS | EMA Stationary Engine Committee to advise Aerotherm of suggestions for a definition of engines to be affected by NSPS, engine manufacturer's comments on EPA's mobile source test procedures, and influence of ambient conditions on emission measurements. (Response received 3/21/77.) |

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[a] All meetings held in EPA offices, Durham, NC unless otherwise noted.

[b] Many other telephone conversations occurred during which procedural details were discussed. These are not documented here.

## TABLE A-4. Continued

| Date | Format[a] | Purpose | Conclusions/Redirections |
|---|---|---|---|
| 2/24/77 | EPA-Aerotherm meeting; report prepared by Aerotherm (3/1/77) | Status report on revisions to SSEIS | EPA to meet with Aerotherm to discuss alternatives for determining an emission standard |
| 3/9-10/77 | EPA-Aerotherm meeting at Mountain View, CA ; report prepared by Aerotherm (3/15/77) | To review emission data compilation and to discuss the development of alternative control strategies | Controlled levels to be determined by applying a percentage reduction demonstrated by viable control technologies to sales-weighted uncontrolled averages |
| 3/24/77 | Letter (EPA Project Officer to Aerotherm) | Advise Aerotherm of format revised draft SSEIS | Aerotherm provided EPA with tables of contents for chapters of SSEIS undergoing major revision (letter of 4/15/77) |
| 9/21-22/77 | EPA-Aerotherm meeting; report prepared by Aerotherm (9/28/77) | To review revisions being made to draft SSEIS; in particular, the alternative control techniques and the definition of affected facilities | Aerotherm to work with EPA contractor in revising economic impact analysis. Aerotherm to deliver revised draft SSEIS by November 1977, economic impact analysis to be completed in December 1977. |
| 11/15,16/77 | EPA-Aerotherm meeting at Mountain View CA; report prepared by Aerotherm (11/21/77) | To discuss change in Aerotherm's scope of work and review draft SSEIS | Aerotherm to write rationale chapter, preamble and regulation with assistance from EPA; Chapters 3, 4, 5, 6, 7, and Appendices A and C approved. |

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[a] All meetings held in EPA offices, Durham, NC unless otherwise noted.

[b] Many other telephone conversations occurred during which procedural details were discussed. These are not documented here.

TABLE A-4. Concluded

| Date | Format[a] | Purpose | Conclusions/Redirections |
|---|---|---|---|
| 1/25/78 | EPA-Aerotherm meeting to discuss economics impact chapter | General discussion of development of economics impact chapters and discussion of future schedule | Future schedule to prepare for NAPCTAC meeting defined |
| 4/5-6/78 | EPA-Aerotherm meeting at OAQPS and Acurex presentation to NAPCTAC | Prepared for NAPCTAC presentation to be given the following day in Washington | Engine manufacturers presented viewpoints and requested future meeting to discuss problem areas |
| 4/26-29/78 | EPA-Aerotherm manufacturers meeting to discuss questions arising from NAPCTAC | Receive from manufacturers their comments and viewpoints on standards recommended at NAPCTAC | Letter drafted and sent to manufacturers specifically requesting additional information to help reevaluate suggested standards |
| 5/30/78 | EPA-Aerotherm meeting to discuss rationale chapter | To discuss organization of rationale chapter and modification of proposed standards based on data received from manufacturers | Standards to be revised based on operational data supplied by manufacturers and statistical analysis of impact of standard |
| 6/29-30/78 | EPA-Aerotherm meeting on rationale, preamble regulation | Discuss final revisions to rationale chapter, preamble and regulation-revision to test method and monitoring discussed | Set schedule for delivery of draft of SSEIS to Steering Committee test method to be based on Method 20 |

[a] All meetings held in EPA offices, Durham, NC unless otherwise noted.

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[b] Many other telephone conversations occurred during which procedural details were discussed. These are not documented here.

## REFERENCES FOR APPENDIX A

(1) Diesel and Gas Turbine Worldwide Catalog. Diesel and Gas Turbine Progress. Milwaukee, Wisconsin. 39. 1974.

(2) McGowin, C. G., Stationary Internal Combustion Engines in the United States. EPA-R2-73-210. April 1973.

(3) Roessler, W. V., A. Muraszew, and R. D. Kopa. Assessment of the Applicability of Automotive Emission Control Technology to Stationary Engines. EPA-650/2-74-051. July 1974.

(4) Offen, G. R. (Acurex/Aerotherm). Trip Report. San Jose Water Pollution Control Plant. (Interoffice Memorandum to F. E. Moreno, Acurex/Aerotherm) June 5, 1975.

# APPENDIX B

| Agency Guidelines for Preparing Regulatory Action Environmental Impact Statements (39 FR 37419) | Location Within the Standards Support and Environmental Impact Statement |
|---|---|
| 1. Background and Description of Proposed Action | |
| Summary of Proposed Standards | The standards are summarized in Chapter 1. |
| Statutory Basis for the Standard | The statutory basis for the standard is given in Chapter 2. |
| Facility Affected | A description of the facility to be affected is given in Chapter 3. |
| Process Affected | A description of the process to be affected is given in Chapter 3. |
| Availability of Control | Information on the availability of control technology is given in Chapter 4. |
| Existing Regulations at State or Local Level | A discussion of existing regulations on the industry to be affected by the standard is included in Chapter 3, Section 2.3. |
| 2. Alternatives to the Proposed Action | |
| Environmental Impacts | Environmental effects of delaying the standards are discussed in Chapter 7, Section 6. |
| Costs | The costs of alternative control techniques are discussed in Chapter 8, Sections 2, 3, and 4. |

| Agency Guidelines for Preparing Regulatory Action Environmental Impact Statements (39 FR 37419) | Location Within The Standards Support and Environmental Impact Statement |
|---|---|
| 3. Environmental Impact of Proposed Action | |
| Air Pollution | The air pollution impact of the standards is considered in Chapter 7, Section 1. |
| Water Pollution | The water pollution impact of the standards is discussed in Chapter 7, Section 2. |
| Solid Waste Disposal | The solid waste disposal impact of the standards is discussed in Chapter 7, Section 3. |
| Energy | The energy impact of the standards is considered in Chapter 7, Section 4. |
| Other | The environmental impacts related to noise and thermal pollution are discussed in Chapter 7, Section 5. |

# APPENDIX C

# EMISSION SOURCE TEST DATA

## C.1 DATA

This appendix tabulates all the quantitative emission data that were obtained for large-bore engines during the preparation of the Standard Support and Environmental Impact Statement. These data include emission and fuel consumption rates for both baseline and controlled conditions. Since no controlled stationary reciprocating engines are known to exist in field installations, all the data were obtained from engine manufacturers, who reported on the results of experiments in their laboratories. In some cases, these data have been made public, but in most instances they were received directly from the manufacturer as a private communication.

Large stationary engines are expensive to purchase and operate, and controlling their emissions has not previously received high priority. For these reasons, virtually no government, commercial (other than manufacturers), or university test labs have obtained one or reported on possible emission reduction technologies that might apply to currently produced engines.[1] However, Southwest Research Institute has a two-cylinder version of an Electromotive Division (EMD) locomotive engine which they have used for emission control investigations(1). This engine is no longer being produced by EMD, although it represents most of the current locomotive engine population. According to data supplied by EMD, this model's successor has higher baseline $NO_x$ emissions (14.7 g/hp-hr for the newer model compared to 9.1 g/hp-hr for the older two-cylinder version at rated conditions) and lower total hydrocarbon emissions (0.49 g/hp-hr compared to 0.92 g/hp-hr). These differences are consistent with the increased efficiency of the engine. New EMD data (Engines 17, 18 and 19) showed the same trends with the application of controls that the SwRI engine data did; consequently, SwRI's results are not included here.

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[1] Recently attained improvement in thermal efficiency by operation at increased pressure and temperature precludes the use of emissions data from older engines.

The data include engines which are currently marketed for stationary application and also several units used in mobile and marine applications. However, data from installed engines were excluded because these older units are not the same as those new units that would have to meet any promulgated standards.

$NO_x$ emissions were measured using one of the four procedures and several kinds of $NO_x$ analyzers. These procedures and instruments are identified and discussed in Section 4.2 for each large-bore engine manufacturer that reported emissions data. $NO_x$ emissions were measured by either a nondispersive analyzer or a chemiluminescent instrument with a thermal reactor to convert the $NO_2$ to NO. Results were reported as grams $NO_2$.

Nondispersive infrared analyzers (NDIR) were used to measure the CO concentrations. Since the exhaust was "dry," interference due to water was negligible. When $CO_2$ and oxygen were measured, an NDIR analyzer was used for the $CO_2$ and a paramagnetic analyzer for the oxygen. These constituents were generally sampled to check the other measurements or to calculate total exhaust flowrate if the inlet air flowrate into the engine was not measured.

Total hydrocarbon emissions were measured in a heated flame ionization detector (FID). The tubing and FID were maintained at 375°F to prevent condensation, and hence, removal of any of the heavier hydrocarbons. Therefore, these measurements were made on a "wet" basis because the water had not been removed. The results were corrected for the water content to make them correspond with the $NO_x$ and CO data. Since the exhaust from gas engines does not contain these easily condensible heavy hydrocarbons, the lines and FID were not heated in some tests on gas engines. In addition to these total hydrocarbon measurements, a few manufacturers measured the nonmethane emissions from gas and dual-fuel engines. Their methods and results are discussed in Appendix C.4.

Four engine manufacturers (Colt, Cooper, DeLaval, and White Superior), who are members of the Diesel Engine Manufacturers Association (DEMA), measured emissions data according to the measurement procedure published by DEMA(2). Although the DEMA methodology suggests that a three-mode cycle be used for engines that will be used at constant speed (essentially all stationary applications) and a six-mode cycle for units that will see variable speed duty (marine and locomotive), most of the results reported here were collected prior to the final publication of the DEMA procedures. Hence, emissions were usually measured only at rated load and speed. However, some engines were also tested at selected part load and speed points. These additional data points appear on the following tables (see Tables C-1 through C-82) as derated operating points or as variations in the engine operating speed.

Several of the engines (Engine Nos. 30 through 33) at the low horsepower end of the large engine category were tested according to the

California 13-mode cycle. However, only the emissions at rated conditions were used here since the 13-mode automotive cycle does not represent stationary engine usage, and no relationship was found between emission rates based on the average of the California 13-mode cycle and those at rated conditions.

The data were submitted in either tabular or graphical form. In the latter case the data points were clearly marked and could be converted readily to the tabular format for inclusion in this appendix. Other operational data were either supplied by the manufacturer or obtained from product information literature. Fuel consumption values in all cases are specific to the test that measured the emissions and are not from general product specifications.

Fuel consumption was generally reported in terms of lb/hp-hr for diesels and Btu/hp-hr for gas and dual-fuel engines. Conversion to kcal/hp-hr was accomplished using a lower heating value of 18,320 Btu/lb for diesel fuel and a units conversion of 0.252 kcal/Btu. All data presented in this document on emissions from diesel and dual-fuel engines are based on operation with No. 2 distillate (diesel oil).

In several cases, a number of tests were performed on the same engine, and consequently an uncontrolled value was reported for each control technique. In all cases the uncontrolled level precedes any controlled data reported for a particular test. In the tabulations which follows, a separate table is used for each engine, and the data are printed in numerical order based on engine code number. These data come from References 1 through 22. These references are noted on the tables except where an engine manufacturer has requested anonymity.

## TABLE C-1. EMISSIONS SOURCE TEST DATA -- ENGINE 1

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 1 | Gas | 82.5 bmep (psi) | Uncontrolled | 15.1 | 0.3 | 1.9 | -- | 7127 | -- | 62.5 | -- | -- | -- |
| | | 1080 hp | Derate power 12% | 3.84 | 0.72 | 2.68 | -- | 7311 | 2.6 | -- | -- | -- | -- |
| | | 300 rpm | Air-to-fuel ratio inc. 25% | 14.66 | 0.22 | 2.14 | -- | 7223 | 1.3 | 68.2 | -- | -- | -- |
| | | 2 cycle, injected | Inc. to 330 rpm at constant power | 6.41 | 0.41 | 2.25 | -- | 7192 | 0.9 | 68.0 | -- | -- | -- |
| | | Blower scavenged BS (130°F) | Inc. to 400 rpm at constant power | 1.25 | 1.45 | 0.87 | -- | 8291 | 16.3 | -- | -- | -- | -- |
| | | 8 cylinder | Dec. intake manifold air temperature 30°F | 9.6 | 0.34 | 2.04 | -- | 7112 | -0.2 | 61.9 | -- | -- | -- |
| | | 14" bore x 14" stroke<br>2115 in³/cyl | Inc. exhaust back-pressure | 9.53 | 0.3 | 2.16 | -- | 7673 | 7.7 | -- | -- | -- | -- |
| | | 135 hp/cyl | 3° ignition retard | 14.25 | 0.35 | 2.0 | -- | 7378 | 3.5 | -- | -- | -- | -- |
| | | Cooper-Bessemer Company | 6° ignition retard | 12.75 | 0.35 | 2.26 | -- | 7496 | 5.2 | 59.0 | | | |
| | | | 9° ignition retard | 13.5 | 0.35 | 2.0 | -- | 7751 | 8.7 | -- | -- | -- | -- |
| | | Model GMVA-8<br>Ref. 3, 4 | 6° ignition retard and decrease intake manifold air temp. 30°F | 10.63 | 0.32 | 2.08 | -- | 7572 | 6.2 | 58.8 | -- | -- | -- |
| | | | Above plus air-to-fuel ratio 13% inc. | 8.73 | 0.31 | 2.19 | -- | 7654 | 7.4 | 66.2 | -- | -- | -- |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-2. EMISSIONS SOURCE TEST DATA -- ENGINE 2

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 2 | Gas | 111.4 bmep<br>1600 hp<br>330 rpm<br>2 cycle injected<br>Turbocharged<br>Aftercooled (130°F)<br>8 cylinder<br>14" bore x 14" stroke<br>2155 in³/cyl<br>200 hp/cyl<br>Cooper-Bessemer Company<br>Model GMVH-8<br>Ref. 3 | Uncontrolled | 13.42 | 0.2 | 1.45 | -- | 6775 | -- | 54.7 | -- | -- | -- |
| | | | Derate power 25% | 4.59 | 0.3 | 1.76 | -- | 7186 | 6.1 | 51.2 | -- | -- | -- |
| | | | 4° ignition retard | 11.76 | 0.27 | 1.47 | -- | 7069 | 4.3 | 52.1 | -- | -- | -- |
| | | | Inc. air-to-fuel ratio to 10% excess air | 7.43 | 0.22 | 1.69 | -- | 6873 | 1.4 | 59.0 | -- | -- | -- |
| | | | Dec. intake manifold air temperature 30°F | 8.27 | 0.21 | 1.58 | -- | 6810 | 0.5 | 54.8 | -- | -- | -- |
| | | | 4° ignition retard & decision intake manifold air temp. by 30°F | 8.09 | 0.22 | 1.57 | -- | 7137 | 5.3 | 52.2 | -- | -- | -- |
| | | | Above plus inc. air-to-fuel ratio to 10% excess air | 4.59 | 0.22 | 1.63 | -- | 7210 | 6.4 | 56.5 | -- | -- | -- |

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[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-3. EMISSIONS SOURCE TEST DATA -- ENGINE 3

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 3 | Diesel | 141 hp bmep<br>2475 hp<br>900 rpm<br>2 cycle<br>Turbocharged<br>Aftercooled<br>12 cylinders<br>9.06" bore x 10" stroke<br>645 in³/cyl<br>206 hp/cyl<br>ElectroMotive Division, General Motors Corporation<br>Model 21-645E-3<br>Ref. 5 | Uncontrolled[c] | 12.61 | 1.19 | 0.38 | -- | -- | -- | -- | -- | -- | -- |

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[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Rated load

## TABLE C-4. EMISSIONS SOURCE TEST DATA -- ENGINE 4

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 4 | Gas | 175 bmep | Uncontrolled | 12.75 | -- | 4.26 | -- | 6374 | -- | 32.5 | -- | -- | -- |
|  |  | 3240 hp | Derate power 14% | 12.14 | -- | 3.38 | -- | 6426 | 0.8 | 31.1 | -- | -- | -- |
|  |  | 514 rpm | 4° ignition retard | 12.43 | -- | 2.63 | -- | 6464 | 1.4 | 32.0 | -- | -- | -- |
|  |  | 4 cycle<br>Turbocharged | Inc. air-to-fuel ratio to 10% excess air | 8.57 | -- | 7.3 | -- | 6520 | 2.3 | 34.7 | -- | -- | -- |
|  |  | Aftercooled (to 100°F) | Dec. intake manifold air temp. by 20°F | 11.62 | -- | 5.27 | -- | 6372 | -0.03 | 32.7 | -- | -- | -- |
|  |  | 12 cylinder<br>13.5" bore x 16.5" stroke<br>2361 in³/cyl | 4° ignition retard & inc. air-to-fuel ratio to 10% excess air | 8.18 | -- | 4.76 | -- | 6593 | 3.4 | 33.8 | -- | -- | -- |
|  |  | 270 hp/cyl<br>Cooper-Bessemer Company<br>Model KSV-12<br>Ref. 3 | Above plus dec. in intake manifold air temp. by 20°F | 7.5 | -- | 4.76 | -- | 6600 | 3.5 | 33.8 | -- | -- | -- |

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[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Smoke measured with a Bacharach Model RXB smoke meter (filter-type instrument)

## TABLE C-5. EMISSIONS SOURCE TEST DATA -- ENGINE 5

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 5 | Diesel | 200 bmep | Uncontrolled | 10.99 | 3.85 | 0.13 | 4 | 6677 | — | 32.6 | 85 | 100 | — |
| | | 4300 hp | Derate power 38% | 10.30 | 1.33 | 0.32 | 2 | 7000 | 4.8 | — | — | — | — |
| | | 600 rpm | 4° ignition retard | 9.31 | 4.45 | 0.21 | 4 | 6732 | 0.8 | 33.1 | 100 | 100 | — |
| | | 4 cycle<br>Turbocharged | Air-to-fuel ratio with 2% excess air | 11.05 | 3.06 | 0.18 | 4 | 6596 | -1.2 | 33.7 | — | — | — |
| | | Aftercooled (to 130°F) | Dec. intake manifold air temp. by 30°F | 10.54 | 3.82 | 0.17 | 4 | 6583 | -1.4 | 33.1 | — | 70 | — |
| | | 12 cylinder<br>13.5" bore x 16.5" stroke | Induct water at 0.17 water-to-fuel ratio | 10.2 | 4.25 | 0.16 | 4 | 6643 | -0.5 | 32.5 | — | — | — |
| | | 2361 $in^3$/cyl<br>358 hp/cyl | 4° ignition retard dec. intake manifold air temp. by 30°F | 8.71 | 4.09 | 0.2 | 4 | 6636 | -0.6 | 33.7 | 116 | 70 | — |
| | | Cooper-Bessemer Company<br>Model KSV-12 | Above plus leaner air-to-fuel ratio using 6% excess air | 8.66 | 3.39 | 0.2 | 4 | 6583 | -1.4 | 35.2 | — | — | — |
| | | Ref. 3, 6 | Above plus water induction at 0.21 water-to-fuel ratio | 7.98 | 3.67 | 0.2 | 4 | 6664 | -0.2 | 34.8 | — | — | — |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-6. EMISSIONS SOURCE TEST DATA -- ENGINE 6

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 6 | Dual fuel | 200 bmep<br>4300 hp<br>600 rpm<br>4 cycle<br>Turbocharged<br>Aftercooled (to 130°F)<br>12 cylinder<br>13.5" bore x 16.5" stroke<br>2362 in³/cyl<br>358 hp/cyl<br>Cooper-Bessemer Company<br>Model KSV-12<br>Ref. 3, 6 | Uncontrolled | 8.96 | 4.5 | 5.16 | — | 6340 | — | — | 100 | 100 | 28.77 |
| | | | Derate power 12% | 9.56 | 2.97 | 6.20 | — | 7025 | 10.8 | — | 83 | 100 | 28.77 |
| | | | 4° ignition retard | 8.41 | 7.21 | 3.22 | — | 6410 | 1.1 | — | 88 | 100 | 28.77 |
| | | | Lean air-to-fuel ratio with 10% excess air | 6.70 | 2.00 | 7.82 | — | 6393 | 0.8 | — | 86 | 100 | 28.77 |
| | | | Dec. intake manifold air temp. by 30°F | 7.27 | 4.25 | 5.28 | — | 6377 | 0.6 | — | 86 | 70 | 28.77 |
| | | | Induct water at 0.1 water-to-fuel rate | 8.35 | 3.39 | 6.48 | — | 6374 | 0.5 | — | 94 | 100 | 28.77 |
| | | | 4° ignition retard & dec. intake manifold air temp. by 30°F | 6.63 | 6.45 | 3.2 | — | 6444 | 1.6 | — | 88 | 70 | 28.77 |
| | | | Above plus inc. air-to-fuel ratio to 10% excess air | 5.29 | 3.44 | 7.39 | — | 6530 | 3.0 | — | 81 | 70 | 28.77 |
| | | | Above plus water induction at water-to-fuel ratio of 0.1 | 5.27 | 3.17 | 7.76 | -- | 6568 | 3.6 | — | -- | — | 28.77 |

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[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Operated with minimum pilot oil recommended by manufacturer

## TABLE C-7. EMISSIONS SOURCE TEST DATA -- ENGINE 7

1-696

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 7 | Dual fuel[c] | Large, low-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 $in^3$/cyl | Uncontrolled | 12.7 | 1.35 | 0.85/<br>0.2 | -- | 6400 | — | 33 | 20 | 38 | 29.03 |
| | | | Derate power 18% | 10.55 | 1.7 | 1.05 | -- | 6600 | 3.1 | 32 | 27 | 44 | 29.02 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Operated with minimum pilot oil recommended by manufacturer

## TABLE C-8. EMISSIONS SOURCE TEST DATA -- ENGINE 8

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 8 | Diesel | Repeat from Engine #7 | Uncontrolled | 10.8 | 1.71 | 1.0 | 6 | 6643 | — | 35 | 16 | 68 | 29.09 |
| | | | 4° ignition retard | 9.1 | — | — | 7.5 | 6881 | 3.6 | 35 | 16 | 68 | 29.09 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Smoke measured with Bosch meter

## TABLE C-9. EMISSIONS SOURCE TEST DATA -- ENGINE 9

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 9 | Diesel | Large, low-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in$^3$/cyl | Uncontrolled | 11.4 | -- | 0.44 | 6 | 6643 | -- | -- | -- | -- | -- |
|  |  |  | 4° ignition retard | 9.4 | -- | 0.44 | 7.5 | 6817 | 2.6 | -- | -- | -- | -- |
|  |  |  | Reduce compression ratio | 10.6 | -- | -- | 6.3 | 6936 | 4.4 | -- | -- | -- | -- |

T-666

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Smoke measured with Bosch meter

## TABLE C-10. EMISSIONS SOURCE TEST DATA -- ENGINE 10

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, in. Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 10 | Diesel | Large, medium-speed, 2-cycle, blower scavanged engine. Greater than 350 in³/cyl | Uncontrolled | 18.8 | 0.63 | 0.55 | 8 | 7844 | -- | 40 | 38 | 68 | 28.93 |
|  |  |  | Derate power 25% | 14.5 | 0.76 | 0.28 | 4 | 8046 | 2.6 | 51 | 38 | 69 | 28.93 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Smoke measured with Bosch meter

## TABLE C-11. EMISSIONS SOURCE TEST DATA -- ENGINE 11

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 11 | Diesel | Large, medium-speed, 2-cycle, blower scavenged engine. Greater than 350 in³/cyl | Uncontrolled | 16.3 | 0.53 | 0.18 | 7.5 | 6951 | — | 45.8 | 47 | 80 | 29.02 |
| | | | Derate power 5% | 12.9 | 0.64 | 0.27 | 3.6 | 7234 | 4.1 | 57.3 | 50 | 80 | 29.02 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Smoke measured with Bosch meter

## TABLE C-12. EMISSIONS SOURCE TEST DATA -- ENGINE 12

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 12 | Diesel | Large, medium speed, 2-cycle turbocharged, aftercooled engine. Greater than 350 in³/cyl | Uncontrolled | 10.7 | 0.85 | 0.16 | -- | 6679 | -- | 37 | 28 | 70 | 29.44 |
| | | | 2° ignition retard | 9.8 | 1.0 | 0.14 | -- | 6753 | 1.1 | -- | -- | -- | -- |
| | | | 4° ignition retard | 8.6 | 1.15 | 0.14 | -- | 6845 | 2.5 | 37 | 31 | 70 | 29.43 |
| | | | 6° ignition retard | 7.3 | 1.2 | 0.2 | 10.7 | 6936 | 3.8 | -- | -- | -- | -- |
| | | | 8° ignition retard | 6.4 | 1.4 | 0.14 | -- | 7046 | 5.5 | -- | -- | -- | -- |
| | | | 10° ignition retard | 6.0 | 1.5 | 0.14 | -- | 7175 | 7.4 | 40 | 33 | 70 | 29.41 |
| | | | Derate power 25% | 9.9 | 0.88 | 0.14 | 9 | -- | -- | 40 | 22 | 70 | 29.34 |
| | | | 5% exhaust gas recirculation | 9.7 | 1.2 | 0.15 | 10.0 | 6716 | 0.5 | 36 | 34 | 70 | 29.03 |
| | | | 12% exhaust gas | 7.1 | 1.2 | 0.15 | 14.8 | 6716 | 0.5 | 33 | 34 | 70 | 29.03 |

T-656

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[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Smoke measured with Bosch meter

## TABLE C-13. EMISSIONS SOURCE TEST DATA -- ENGINE 13

T-886

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 13 | Dual Fuel | Large medium speed, 2 cycle, turbocharged, aftercooled engine. Greater than 350 in³/cyl | Uncontrolled | 8.8 | 1.71 | 1.54/<br>0.35 | — | 7040 | — | 31 | 40 | 70 | 28.92 |
| | | | 2° ignition retard | 8.41 | 1.85 | 1.42 | — | 7182 | 2.0 | 30 | 40 | 70 | 28.92 |
| | | | 6° ignition retard | 6.7 | 1.98 | 1.47 | — | 7284 | 3.5 | 31 | 44 | 70 | 28.95 |
| | | | 8° ignition retard | 2.6 | 2.18 | 1.76 | — | 7753 | 10.1 | 34 | 42 | 70 | 28.95 |
| | | | Lean air-to-fuel ratio by 17% | 6.73 | 1.86 | 1.84 | — | 7220 | 2.5 | 36 | 37 | 70 | 28.92 |
| | | | Dec. manifold air temperature by 20°F (from 120°F) | 5.0 | 0.5 | 2.4 | — | 7075 | 0.5 | — | — | 70 | 29.00 |
| | | | 10.7% exhaust gas recirculation | 6.6 | 1.66 | 1.41 | — | 7100 | 0.8 | 27 | 56 | 70 | 29.02 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Operated with minimum pilot oil recommended by manufacturer

## TABLE C-14. EMISSIONS SOURCE TEST DATA -- ENGINE 14

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 14 | Dual Fuel[c] | Large, medium speed, 2 cycle, turbocharged, aftercooled engine. Greater than 350 $in^3$/cyl | Uncontrolled | 14.93 | 1.25 | 2.66 | -- | 7170 | -- | 36 | 15 | 70 | 29.15 |
| | | | Inc. speed 25% at constant torque | 8.8 | 1.71 | 1.54 | -- | 7040 | -1.8 | 31 | 40 | 70 | 28.92 |

T-686

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[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Operated with minimum pilot oil recommended by manufacturer

## TABLE C-15. EMISSIONS SOURCE TEST DATA -- ENGINE 15

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 15 | Diesel | 150 bmep | Uncontrolled | 11.24 | 3.02 | 0.46 | Bosch #3 | 6366 | -- | 31.9 | 69 | 73 |  |
|  |  | 2410 hp | 4° ignition retard | 10.01 | 4.44 | 0.40 | Bosch #4 | 6459 | 1.5 | 31.3 | 52 | 84 |  |
|  |  | 450 rpm<br>4 cycle | Inc. speed to 500 rpm, const. torque | 12.07 | 2.39 | 0.43 | Bosch #3 | 6250 | -1.8 | -- | 58 | 91 |  |
|  |  | Turbocharged | Decrease air-to-fuel ratio 12% | 11.91 | 4.65 | 0.26 | -- | 6522 | 2.5 | 28.2 | 61 | 68 |  |
|  |  | Aftercooled (to 90°F) | Derate 27% | 10.4 | 1.56 | 0.48 | Bosch #2 | 6320 | -0.7 | 36.8 | 63 | -- |  |
|  |  | 12 cylinder | Inc. manifold air temp. by 30°F | 12.24 | 1.03 | 0.42 | -- | 6417 | 0.8 | -- | 67 | 72 |  |
|  |  | 13.5" bore x 16.5" stroke | Dec. manifold air temp. by 20°F | 11.54 | 0.95 | 0.45 | -- | 6294 | -1.1 | 32.3 | 66 | 72 |  |
|  |  | 2361 in³/cyl | Inc. valve overlap[c] | 12.29 | 1.07 | 0.45 | Bosch #4 | 6367 | 0.02 | 32.0 | 56 | 67 |  |
|  |  | 200 hp/cyl<br>Cooper-Bessemer Company | Induct water at 0.21 water-to-fuel ratio | 10.74 | 2.95 | 0.51 | -- | 6351 | -0.2 | 31.8 | 42 | 70 |  |
|  |  | Model KSV-12-GDT | Induct water at 0.58 water-to-fuel ratio | 9.6 | 2.99 | 0.41 | -- | 6407 | 0.6 | 31.7 | 67 | 71 |  |
|  |  | Ref. 8 | 4° ignition retard & 20°F dec. intake manifold air temp. | 9.44 | 3.84 | 0.39 | -- | 6514 | 2.3 | 31.0 | 59 | 87 |  |
|  |  |  | Above plus water induction at 0.21 water-to-fuel rate | 8.0 | 3.62 | 0.38 | -- | 6496 | 2.0 | 31.3 | 56 | 75 |  |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Degree of overlap not noted

## TABLE C-16. EMISSIONS SOURCE TEST DATA -- ENGINE 16

T-666

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 16 | Dual fuel | 150 bmep | Uncontrolled | 7.79 | 3.69 | 5.67 | Bosch #2 | 6464 | -- | 34.3 | 44 | 80 |  |
|  |  | 2410 hp | 2° ignition retard | 6.67 | -- | -- | -- | 6655 | 2.9 | — | 45 | 81 |  |
|  |  | 450 rpm | 3° ignition retard | 6.31 | 4.78 | 5.08 | Bosch #2 | 6655 | 2.9 | 33.7 | 64 | 91 |  |
|  |  | 4 cycle<br>Turbocharged | Inc. air-to-fuel ratio to 10% excess air (at same ratio of pilot oil-to-gas) | 5.92 | 2.02 | 7.94 | Bosch #2 | 6648 | 2.8 | 35.7 | 61 | 80 |  |
|  |  | Aftercooled (to 90°F) | Inc. rpm to 500 at constant torque | 6.42 | 4.16 | 5.35 | Bosch #3 | 6571 | 1.6 | — | 69 | 92 |  |
|  |  | 12 cylinder<br>13.5" bore x 16.5" stroke | Inc. inlet manifold air temp. by 30°F | 9.53 | 3.24 | 6.11 | -- | 6516 | 0.8 | -- | 62 | 82 |  |
|  |  | 2361 in³/cyl | Dec. intake manifold air temp. by 20°F | 6.58 | 3.81 | 6.34 | -- | 6614 | 2.3 | 33.5 | 46 | 84 |  |
|  |  | 200 hp/cyl | Induct water at 0.25 water-to-fuel rate | 6.90 | 3.55 | 6.73 | -- | 6638 | 2.7 | 33.0 | 63 | 93 |  |
|  |  | Cooper-Bessemer Company | Inc. valve overlap by 16°, 3° retard | 7.34 | 4.30 | 6.07 | Bosch #2 | 6765 | 4.6 | -- | 56 | 88 |  |
|  |  | Model KSV-12-GDT<br>Ref. 8 | 3° ignition retard & dec. intake manifold air temp. by 20°F | 5.88 | 8.62 | 3.03 | -- | 6760 | 4.6 | 32.6 | 47 | 91 |  |
|  |  |  | Above plus inc. air-to-fuel ratio by 10% excess air | 5.17 | 4.06 | 6.38 | -- | 6967 | 7.8 | 34.3 | 47 | 90 |  |
|  |  |  | Above plus water induction at water-to-fuel ratio of 0.2 | 4.63 | 8.35 | 4.06 | -- | 7255 | 12.2 | 33.8 | 46 | 84 |  |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-17. EMISSIONS SOURCE TEST DATA -- ENGINE 17

T-666

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr[c] | | | Visible Emission, Percent Opacity[d] | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 17 | Diesel | 94 bmep<br>2200 hp | Uncontrolled | 14.6 | 5.4 | 0.19 | — | — | | | | | |
| | | 900 rpm<br>2 cycle | 2° ignition retard (min sac injector) | 12.49 | 6.03 | 0.18 | — | — | | | | | |
| | | Blower scavenge (100°F inlet air) | 4° ignition retard (min sac injector) | 10.57 | 6.75 | 0.19 | — | — | | | | | |
| | | | Uncontrolled | 18.59 | 2.56 | 0.34 | — | 7478 | | | | | |
| | | 16 cylinder | Increased port width | 16.09 | 2.35 | 0.34 | — | 7386 | | | | | |
| | | 9.06" bore x 10" stroke | Uncontrolled | 19.09 | 3.49 | 0.33 | — | — | | | | | |
| | | 645 in³/cyl<br>138 hp/cyl | Dec. inlet manifold air temp. 25°F (to 100°F) | 18.6 | 1.87 | 0.35 | — | — | | | | | |
| | | ElectroMotive Division, General Motors Corporation | Uncontrolled<br>⋮ | 12.56 | 8.36 | 0.80 | 8.5 | — | | | | | |
| | | | Install low sac injectors | 12.32 | 7.28 | 0.64 | 4.5 | — | | | | | |
| | | Model 16-645E | Baseline (low sac) | 14.96 | 6.28 | 0.49 | 4.5 | — | | | | | |
| | | Ref. 5 | Install minimum sac injectors | 14.6 | 5.4 | 0.19 | — | — | | | | | |
| | | | Uncontrolled | 15.03 | 5.10 | 0.49 | — | 7651 | | | | | |
| | | | Induct water at water-to-fuel ratio of 0.53 | 12.36 | 3.71 | 0.50 | — | 7580 | | | | | |
| | | | (Continued) | | | | | | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] All data EMD 10 mode composite except retard which is rated load

[d] Smoke measured with FHS (light extinction) meter. Also Engines 18, 19, 11-3/4" exhaust stack diameter.

# TABLE C-17. Concluded

| Engine Code No. | Fuel | Description | Control | | Emissions, g/hp-hr[c] | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 17 | Diesel | 94 bmep | Uncontrolled, rated load | | 15.67 | 2.53 | 0.55 | -- | -- | | | | | |
| | | 2200 hp | | | | | | | | | | | | |
| | | 900 rpm | Uncontrolled emissions, statistical sample | -3σ | 13.27 | 0.95 | 0.35 | 3 | -- | Average of 13 engines Data is EMD composite cycle, 3 variation avg. of 13 engines. Inlet air temperature variation 72° to 107°. | | | | |
| | | | | avg | 16.00 | 2.39 | 0.62 | 3 | -- | | | | | |
| | | | | +3σ | 18.73 | 3.83 | 0.89 | 3 | -- | | | | | |
| | | 2 cycle | | | | | | | | | | | | |
| | | Blower scavenge (100°F inlet air) | | | | | | | | | | | | |
| | | 16 cylinder | | | | | | | | | | | | |
| | | 9.06" bore x 10" stroke | | | | | | | | | | | | |
| | | 645 in³/cyl<br>138 hp/cyl | | | | | | | | | | | | |
| | | ElectroMotive Division, General Motors Corporation | | | | | | | | | | | | |
| | | Model 16-645E | | | | | | | | | | | | |
| | | Ref. 5 | | | | | | | | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-18. EMISSIONS SOURCE TEST DATA -- ENGINE 18

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 18 | Diesel | 140 bmep | Uncontrolled[c] | 14.74 | 1.23 | 0.48 | -- | 6734 | - |  |  |  |  |
|  |  | 3300 hp | 2° ignition retard | 12.63 | 1.43 | 0.43 | -- | 6826 | 1.4 |  |  |  |  |
|  |  | 900 rpm | 4° ignition retard | 11.36 | 1.53 | 0.38 | -- | 6863 | 1.9 |  |  |  |  |
|  |  | 2 cycle | Uncontrolled[c] | 14.3 | 1.05 | 0.36 | -- | 6753 | - |  |  |  |  |
|  |  | Turbocharged<br>Aftercooled | Dec. manifold air temp. by 40°F (from 101 to 59°F) | 15.3 | 0.5 | 0.4 | -- | 6643 | -1.6 |  |  |  |  |
|  |  | 16 cylinder | Uncontrolled | 12.70 | 1.69 | 1.57 | -- | -- |  |  |  |  |  |
|  |  | 9.06" bore x 10" stroke | Dec. manifold air temp. by 50°F (from 110 to 60°F) | 14.98 | 0.83 | 0.51 | -- | -- |  |  |  |  |  |
|  |  | 30-1/4" exhaust stack diameter | Uncontrolled (85°F air inlet) | 13.85 | 1.39 | 0.57 | -- | -- |  |  |  |  |  |
|  |  | 645 in³/cyl |  |  |  |  |  |  |  |  |  |  |  |
|  |  | 206 hp/cyl | Install low sac injectors[d] | 14.39 | 1.42 | 0.40 | -- | -- |  |  |  |  |  |
|  |  | ElectroMotive Division, General Motors Corporation | Install experimental min. sac volume injectors (85°F inlet air)[d] | 13.79 | 1.57 | 0.38 | -- | -- |  |  |  |  |  |
|  |  | Model 16-645E-3 | Uncontrolled, rated load | 12.37 | 1.23 | .41 | -- | -- |  |  |  |  |  |
|  |  | Ref. 5 | Uncontrolled emisions, statistical sample -3σ<br>avg<br>+3σ | 9.97<br>13.12<br>16.27 | 0.92<br>0.55<br>2.18 | 0.16<br>0.55<br>0.94 | 4<br>4<br>4 | --<br>--<br>-- | Data is EMD composite cycle, 3σ variation, avg. of 10 engines. Inlet air temp. varied from 72° to 107°F. |  |  |  |  |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Rated load

[d] EMD composite

## TABLE C-19. EMISSIONS SOURCE TEST DATA -- ENGINE 19

T-644

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 19 | Diesel | 133 bmep<br>3900 hp<br>900 rpm<br>2 cycle<br>Turbocharged<br>Aftercooled (100°F)<br>20 cylinders<br>9.06" bore x 10" stroke<br>30-1/4" exhaust stack diameter<br>645 $in^3$/cyl<br>195 hp/cyl<br>ElectroMotive Division, General Motors Corporation<br>Model 20-645E-3<br>Ref. 5 | Uncontrolled[c] | 12.99 | 1.56 | 0.59 | -- | -- |  |  |  |  |  |
|  |  |  | Install low sac injectors | 13.07 | 1.23 | 0.30 | -- | -- |  |  |  |  |  |
|  |  |  | Uncontrolled[d], rated load | 11.87 | 0.73 | 0.37 | -- | -- |  |  |  |  |  |
|  |  |  | Uncontrolled emissions, statistical sample<br>$-3\sigma$ avg $+3\sigma$ | 9.36 | 0.51 | 0.30 | 4 | -- | Data is EMD composite cycle. $3\sigma$ variation, avg. of 11 engines. Inlet air temp. varied from 67° to 97°F. |  |  |  |  |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] EMD composite

[d] Average of 11 engines

# TABLE C-20. EMISSIONS SOURCE TEST DATA -- ENGINE 20

T-466

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 20 | Gas | 140 bmep | Uncontrolled | 17.5 | 0.60 | 1.2 | -- | 7300 | -- | -- | -- | 100 | 28.90 |
| | | 2400 hp | 3° ignition retard | 15.24 | 0.62 | 1.19 | -- | 7318 | 0.2 | -- | -- | 100 | 28.90 |
| | | 330 rpm | 5° ignition retard | 13.8 | 0.65 | 1.2 | -- | 7373 | 1.0 | -- | -- | 100 | 28.90 |
| | | 4 cycle | 6° ignition retard | 13.1 | 0.65 | 1.19 | -- | 7391 | 1.2 | -- | -- | 100 | 28.90 |
| | | Turbocharged | 9° ignition retard | 11.25 | 0.70 | 1.19 | -- | 7526 | 3.1 | -- | -- | 100 | 28.90 |
| | | Aftercooled (to 140°F) | 10° ignition retard | 10.2 | 0.71 | 1.19 | -- | 7592 | 4.0 | -- | -- | 100 | 28.90 |
| | | 12 cylinder | Uncontrolled | 18.1 | 0.58 | 1.2 | -- | 7300 | 0.0 | -- | -- | 99 | 28.96 |
| | | 16.25" bore x 18" stroke | Dec. intake manifold air temp. 40°F (to 100°F) | 12.0 | 1.0 | 1.7 | -- | 7264 | -0.5 | -- | -- | 99 | 28.96 |
| | | 3753 in³/cyl | Induct water at 0.92 water-to-fuel ratio | 7.5 | 1.5 | 2.2 | -- | 7375 | 1.0 | -- | -- | 99 | 28.96 |
| | | 200 hp/cyl | | | | | | | | | | | |
| | | Ingersoll-Rand Company | Dec. speed to 280 rpm at constant torque | 21.5 | 0.41 | 1.3 | -- | 7337 | 0.5 | -- | -- | 99 | 28.96 |
| | | | Derate power 15% | 14.5 | 0.90 | 1.28 | -- | 7373 | 1.0 | -- | -- | 100 | 29.22 |
| | | Model 412-KVSR | 25% | 12.5 | 1.18 | 1.60 | -- | 7519 | 3.0 | -- | -- | | |
| | | Ref. 9 | | | | | | | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-21. EMISSIONS SOURCE TEST DATA -- ENGINE 21

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 21 | Gas | 103 bmep | Uncontrolled | 16.6 | 0.67 | -- | -- | 8130 | -- | 17.4 | 75 | 90 | 29.07 |
| | | 400 hp | Retard 50, air-to-fuel change | 11.6 | 11.5 | -- | -- | 8570 | 5.4 | 16.8 | 79 | 89 | 29.27 |
| | | 1000 rpm | Manifold air temp. decrease of 20°F | 16.2 | 0.81 | -- | -- | 8080 | -0.6 | 17.4 | 30 | 70 | 29.27 |
| | | 4 cycles | Manifold air temp. decrease of 20°F and air-to-fuel change | 12.0 | 13.1 | -- | -- | 8230 | 1.2 | 16.9 | 47 | 70 | 29.05 |
| | | Naturally aspirated (95°F inlet) | Derate 25% | 15.5 | 0.52 | -- | -- | 8650 | 6.4 | 17.4 | 82 | 91 | 29.03 |
| | | 6 cylinder | Internal EGR | 15.7 | 0.97 | -- | -- | 8220 | 1.1 | 17.4 | 68 | 89 | 29.17 |
| | | 8.5" bore x 9 stroke | Internal EGR plus air-to-fuel charge | 9.7 | 7.1 | -- | -- | 8470 | 4.2 | 16.7 | 73 | 90 | 29.05 |
| | | 511 in³/cyl | Internal EGR, air-to-fuel charge and $H_2O$ induction | 3.5 | 20.9 | -- | -- | 9070 | 11.6 | 16.5 | 78 | 90 | 29.24 |
| | | 66.7 hp/cyl | | | | | | | | | | | |
| | | White Superior | | | | | | | | | | | |
| | | Model 6G-510 | | | | | | | | | | | |
| | | Ref. 10, 11 | | | | | | | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-22. EMISSIONS SOURCE TEST DATA -- ENGINE 22

T-466

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 22 | Gas | 107 bmep | Uncontrolled | 17.7 | 1.68 | 0.8 | -- | 7750 | -- | 18.8 | 122 | 103 | 28.93 |
|  |  | 800 hp | Derate 25%, A | 11.9 | 8.27 | 1.57 | -- | 8507 | 9.6 | 17.5 | 109 | 103 | 28.95 |
|  |  | 900 rpm | Derate 6%, retard 5°, air-to-fuel change | 11.5 | 7.3[c] | -- | -- | 8130 | 4.8 | 16.1 | 48 | 100 | 29.14 |
|  |  | 4 cycle |  |  |  |  |  |  |  |  |  |  |  |
|  |  | Naturally aspirated (105° F inlet) | Derate 6%, retard 5°, air-to-fuel change, manifold temp. dec. 30°F | 14.7 | 2.7[c] | -- | -- | 8050 | 3.7 | 17.7 | 24 | 65 | 29.14 |
|  |  | 8 cylinder | Decrease speed to 750 rpm, constant torque, air-to-fuel change | 13.1 | 5.1 | 1.21 | -- | 7760 | 0 | 17.4 | 89 | 96 | 28.93 |
|  |  | 10" bore x 10.5" stroke |  |  |  |  |  |  |  |  |  |  |  |
|  |  | 825 in³/cyl |  |  |  |  |  |  |  |  |  |  |  |
|  |  | 100 hy/cyl |  |  |  |  |  |  |  |  |  |  |  |
|  |  | White Superior |  |  |  |  |  |  |  |  |  |  |  |
|  |  | Model 8G-825 |  |  |  |  |  |  |  |  |  |  |  |
|  |  | Ref. 10 |  |  |  |  |  |  |  |  |  |  |  |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-23. EMISSIONS SOURCE TEST DATA -- ENGINE 23

T-466

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 23 | Gas | 147 bmep | Uncontrolled | 22.1 | 0.59 | 6.87 | — | 7440 | — | 26.5 | — | 102 | 28.91 |
| | | 1100 hp | Derate power 25% | 24.2 | 0.52 | 7.91 | — | 7550 | 1.5 | 27.7 | — | 102 | 28.91 |
| | | 900 rpm | Dec. speed to 750 rpm, constant torque | 13.8 | 6.37 | 8.6 | — | 7570 | 1.7 | 26.5 | — | 100 | 28.58 |
| | | 4 cycle | | | | | | | | | | | |
| | | Turbocharged | | | | | | | | | | | |
| | | Aftercooled to 130° | | | | | | | | | | | |
| | | 8 cylinder | | | | | | | | | | | |
| | | 10" bore x 10-1/2" stroke | | | | | | | | | | | |
| | | 825 in³/cyl | | | | | | | | | | | |
| | | 138 hp/cyl | | | | | | | | | | | |
| | | White Superior | | | | | | | | | | | |
| | | Model 8GT-825 | | | | | | | | | | | |
| | | Ref. 10 | | | | | | | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-24. EMISSIONS SOURCE TEST DATA -- ENGINE 24

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 24 | Diesel | 99 bmep | Uncontrolled | 8.7 | 8.5 | 0.25 | 12 | 6930 | -- | 26.5 | -- | 90 | 29.18 |
| | | 610 hp | Derate power 25% | 10.7 | 0.75 | 0.55 | 1 | 6780 | -2.2 | 36.6 | -- | 95 | 29.16 |
| | | 900 rpm | Dec. speed to 750 rpm, constant torque | 8.0 | 4.5 | 0.38 | 6 | 6550 | -5.5 | 38.4 | -- | 89 | 29.13 |
| | | 4 cycle | | | | | | | | | | | |
| | | Naturally aspirated (95°F inlet) | | | | | | | | | | | |
| | | 8 cylinders | | | | | | | | | | | |
| | | 9.125" bore x 10.5" stroke | | | | | | | | | | | |
| | | 687 in³/cyl | | | | | | | | | | | |
| | | 76 hp/cyl | | | | | | | | | | | |
| | | White Superior | | | | | | | | | | | |
| | | Model 40-8 | | | | | | | | | | | |
| | | Ref. 10 | | | | | | | | | | | |

T-656

[a] Total hydrocarbons

[b] Brake specific fuel consumption

# TABLE C-25. EMISSIONS SOURCE TEST DATA -- ENGINE 25

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 25 | Gas | 178 bmep | Uncontrolled | 18.7 | 0.66 | -- | -- | 7200 | -- | 25.7 | 150 | 89 | 29.13 |
| | | 2000 hp | Derate 25% | 7.9 | 0.83 | -- | -- | 7400 | 2.8 | 25.7 | 174 | 103 | 29.12 |
| | | 900 rpm | Dec. speed to 722 rpm at constant torque | 24.8 | 1.21 | 7.5 | -- | 7820 | 8.6 | 20.7 | 157 | 90 | 29.13 |
| | | 4 cycle | | | | | | | | | | | |
| | | Turbocharged | | | | | | | | | | | |
| | | Aftercooled 130°F | | | | | | | | | | | |
| | | 12 cylinders | | | | | | | | | | | |
| | | 10" bore x 10.5" stroke | | | | | | | | | | | |
| | | 825 in³/cyl | | | | | | | | | | | |
| | | 167 hp/cyl | | | | | | | | | | | |
| | | White Superior | | | | | | | | | | | |
| | | Model 12 SGT | | | | | | | | | | | |
| | | Ref. 10 | | | | | | | | | | | |

T-656

[a] Total hydrocarbons

[b] Brake specific fuel consumption

T-686

## TABLE C-26. EMISSIONS SOURCE TEST DATA -- ENGINE 26

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$[c] | CO | $HC_T$[a] | | | | | | | |
| 26 | Diesel | Large, high-speed, 4-cycle, turbocharged engine. Greater than 350 in$^3$/cyl. | Uncontrolled | 8.6 | 2.1 | 0.4 | Bosch #2 | 7147 | -- | -- | -- | -- | -- |
| | | | 4° ignition retard | 6.3 | 2.1 | 0.3 | Bosch #2.4 | 7421 | 3.8 | -- | -- | -- | -- |
| | | | 8° ignition retard | 5.2 | 2.1 | 0.2 | Bosch #2.3 | 7603 | 6.4 | -- | -- | -- | -- |
| | | | Uncontrolled | 9.3 | -- | -- | -- | -- | -- | -- | -- | -- | -- |
| | | | Dec. intake manifold air temp. by 70°F to 105°F | 7.6 | -- | -- | -- | -- | -- | -- | -- | -- | -- |
| | | | Dec. engine speed to 1000 rpm, constant torque | 8.6 | -- | -- | -- | -- | -- | -- | -- | -- | -- |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] NO measured only, reported as $NO_2$

## TABLE C-27. EMISSIONS SOURCE TEST DATA -- ENGINE 27

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$[c] | CO | $HC_T$[a] | | | | | | | |
| 27 | Diesel | High speed, 4 cycle, turbo-charged engine. Bore greater than 350 in³/cyl. | Uncontrolled | 7.4 | 0.8 | 0.5 | Bosch #1.5 | 6667 | -- | | | | |
| | | | Inc. exhaust back pressure by 50" of water | 7.25 | 1.1 | 0.5 | Bosch #2.0 | 6778 | 1.7 | | | | |
| | | | Inc. exhaust back pressure by 100" of water | 7.1 | 1.45 | 0.5 | Bosch #2.5 | 6813 | 2.2 | | | | |
| | | | Derate 25% | 7.1 | 0.65 | 0.2 | -- | -- | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] NO measured only, reported as $NO_2$

## TABLE C-28. EMISSIONS SOURCE TEST DATA -- ENGINE 28

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$[d] | CO | $HC_T$[a] | | | | | | | |
| 28 | Gas | Large, high speed, 4 cycle turbo charged engine. Greater 350 in³/cyl. | Uncontrolled | 12.5 | 9.0 | 0.9 | — | 8139 | — | | | | |
| | | | Inc. intake manifold air temp. by 32°F (to 128°F)[c] | 8.5 | 24.0 | 0.9 | — | 8139 | 0 | | | | |
| | | | Uncontrolled (after cooled to 95°F) | 12.0 | 9.1 | 0.9 | — | 7960 | — | | | | |
| | | | 4° ignition retard | 11.8 | 8.5 | 0.9 | — | 8020 | — | | | | |
| | | | 6° ignition retard | 11.7 | 8.1 | 0.9 | — | 8151 | 0.2 | | | | |
| | | | 4° ignition advance | 11.6 | 10.5 | 0.9 | — | 7929 | — | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Increase in $NO_x$ probably due to change in A/F

[d] NO measured only, reported as $NO_2$

## TABLE C-29. EMISSIONS SOURCE TEST DATA -- ENGINE 29

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$[c] | CO | $HC_T$[a] | | | | | | | |
| 29 | Gas | Large, high speed, 4-cycle, turbocharged engine. Greater than 350 in³/cyl. | Uncontrolled | 13.6 | 18.0 | 1.0 | -- | 7580 | -- | 16.0 | | | |
| | | | Manifold air cooling (aftercooled to 85°F) | 13.5 | 15.0 | 1.0 | -- | 7590 | | | | | |
| | | | Dec. air-to-fuel ratio by 12% (from 16 to 14.1) | 2.0 | 77.0 | 1.5 | -- | 8630 | 13.9 | 14.1 | | | |
| | | | Uncontrolled | 11.85 | 12.05 | 1.6 | -- | -- | -- | | | | |
| | | | Derate 25% | 11.57 | 14.70 | 2.0 | -- | -- | -- | | | | |

---

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] NO measured only, reported as $NO_2$

## TABLE C-30. EMISSIONS SOURCE TEST DATA -- ENGINE 30

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 30 | Diesel | 4 cycle, turbo-charged, medium bore engine | Uncontrolled pre-combustion chamber engine[c] | 6.6[d] | 1.9 | 0.6 | — | — | | | | | |
| | | Aftercooled to 110°F | | 5.3[e] | 0.9 | 0.1 | — | — | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] $NO_x$ corrected to 75 grains humidity as per 38 FR 32138

[d] Based on California 13 mode composite test results

[e] Rated condition of California 13 mode test cycle

## TABLE C-31. EMISSIONS SOURCE TEST DATA -- ENGINE 31

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 31 | Diesel | 4 cycle, turbo-charged medium-bore engine | Uncontrolled pre-combustion chamber engine[c] | 4.6[d] | 4.1 | 0.6 | -- | -- | | | | | |
| | | | | 3.3[e] | 1.5 | 0.1 | -- | -- | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] $NO_x$ corrected to 75 grains humidity as per 38 FR 32138

[d] Based on California 13 mode composite test results

[e] Rated condition of California 13 mode test cycle

## TABLE C-32. EMISSIONS SOURCE TEST DATA -- ENGINE 32

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 32 | Diesel | 4 cycle turbo-charged, medium bore engine | Uncontrolled precombustion chamber engine[c] | 4.0[d] | 1.6 | 0.3 | -- | -- | | | | | |
| | | | | 2.7[e] | 1.0 | 0.1 | -- | -- | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] $NO_x$ corrected to 75 grains humidity as per 38 FR 32138

[d] Based on California 13 mode composite test results

[e] Rated condition of California 13 mode test cycle

## TABLE C-33. EMISSIONS SOURCE TEST DATA -- ENGINE 33

1-666

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 33 | Diesel | 4 cycle turbo-charged, medium bore engine | Uncontrolled precombustion chamber engine[c] | 5.5 | 1.3 | 0.2 |  |  |  |  |  |  |  |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Based on California 13 mode composite test results using intermittent rating as 100% power.

## TABLE C-34. EMISSIONS SOURCE TEST DATA -- ENGINE 34

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 34 | Diesel | 1410 hp<br>1500 rpm<br>4 cycle<br>Turbocharged<br>8 cylinder<br>7.87" bore<br>S.E.M.T. Pielstick<br>Model 8 PA 4200<br>Ref. 14 | Variable throat precombustion chamber[c] | 3.5 | 50 ppm | 100 ppm | Bosch #0.96 | 6869 | | | | | |
| | | | Derate 34% with constant rpm | 3.5 | -- | -- | Bosch #0.90 | 6778 | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Engine operated at rated condition. Emissions measured with procedure equivalent to DEMA test practice

## TABLE C-35. EMISSIONS SOURCE TEST DATA -- ENGINE 35

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 35 | Diesel | Less than 500 hp, higher speed, 4-cycle, naturally aspirated engine. Greater than 350 in³/cyl. | Uncontrolled | 10.0 | 4.3 | 0.3 | -- | 7325 |  |  |  |  |  |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-36. EMISSIONS SOURCE TEST DATA -- ENGINE 36

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 36 | Diesel | Large, high-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in³/cyl. | Uncontrolled | 8.2 | 1.0 | 0.3 | -- | 6778 | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-37. EMISSIONS SOURCE TEST DATA -- ENGINE 37

T-666

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 37 | Gas | Large, high-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in$^3$/cyl. | Uncontrolled | 12.5 | 7.0 | 1.5 | -- | 7500 |  |  |  |  |  |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-38. EMISSIONS SOURCE TEST DATA -- ENGINE 38

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 38 | Gas | Large, high-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in³/cyl. | Uncontrolled 1200 rpm | 12.5 | 11.4 | 1.5 | -- | 8198 | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-39. EMISSIONS SOURCE TEST DATA -- ENGINE 39

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 39 | Gas | Large, high speed, 4-cycle naturally aspirated engine Greater than 350 in³/cyl. | Uncontrolled | 14.0 | 16.0 | 1.0 | -- | 9000 | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-40. EMISSIONS SOURCE TEST DATA -- ENGINE 40

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 40 | Gas | Large, high-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in³/cyl. | Uncontrolled | 7.8 | 29.0 | 1.0 | -- | 8198 | | | | | |

T-656

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[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-41. EMISSIONS SOURCE TEST DATA -- ENGINE 41

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 41 | Diesel | 141 bmep<br>1650 hp<br>900 rpm<br>2 cycle<br>Turbocharged<br>Aftercooled<br>8 cylinders<br>9.06" bore x 10" stroke<br>645 in$^3$/cyl<br>206 hp/cyl<br>ElectroMotive Division, General Motors Corporation<br>Model 8-645E-3<br>Ref. 5 | Uncontrolled | 12.88 | 0.93 | 0.42 | -- | -- | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-42. EMISSIONS SOURCE TEST DATA -- ENGINE 42

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 42 | Diesel | 94 bmep<br>1650 hp<br>900 rpm<br>2 cycle<br>Blower scavenged<br>12 cylinders<br>9.06" bore x 10" stroke<br>645 in³/cyl<br>138 hp/cyl<br>ElectroMotive Division, General Motors Corporation<br>Model 12-645E<br>Ref. 5 | Uncontrolled[c] | 14.74 | 4.14 | 0.49 | — | — |  |  |  |  |  |
|  |  |  | Uncontrolled[d] | 15.52 | 3.65 | 0.57 | — | — |  |  |  |  |  |
|  |  |  | Standard deviation | 0.79 | 0.49 | 0.09 | — | — |  |  |  |  |  |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

[c] Average of 8 engines (rated load)

[d] Data is EMD 10 mode composite cycle, standard deviation, σ, average of 8 engines

## TABLE C-43. EMISSIONS SOURCE TEST DATA -- ENGINE 43

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 43 | Diesel | 1100 hp<br>900 rpm<br>2 cycle<br>Blower scavenged<br>8 cylinders<br>9.06" bore x 10" stroke<br>645 in$^3$/cyl<br>138 hp/cyl<br>ElectroMotive Division, General Motors Corporation<br>Model 8-645E<br>Ref. 5 | Uncontrolled | 17.07 | 2.20 | 0.37 | -- | -- | -- | -- | -- | -- | -- |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-44. EMISSIONS SOURCE TEST DATA -- ENGINE 44

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp. °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 44 | Gas | 80 bmep | Uncontrolled | 29.0 | 0.25 | 0.30 | -- | 7840 | 0 | 21.5 | -- | -- | -- |
| | | 1650 hp | Increase A/F 2% | 23.0 | 0.25 | 0.30 | -- | 7840 | 0 | 22.0 | -- | -- | -- |
| | | 330 rpm | Decrease A/F 21% | 20.5 | 1.65 | 0.35 | -- | 8160 | 4.1 | 17.0 | -- | -- | -- |
| | | 4-cycle, carbureted | Retard 5° | 28.2 | 0.25 | 0.15 | -- | 8600 | 9.7 | 20.0 | -- | -- | -- |
| | | | Retard 9° | 27.9 | 0.25 | 0.08 | -- | 9040 | 15.3 | 19.0 | -- | -- | -- |
| | | Naturally aspirated | Retard 5° & dec. A/F | 14.0 | 8.4 | 0.35 | -- | 8720 | 11.2 | 16.6 | -- | -- | -- |
| | | 12 cylinder | | | | | | | | | | | |
| | | 16.25" bore x 18" stroke | Inc. speed to 380 rpm | 25.0 | 0.30 | 0.25 | -- | 8240 | 5.1 | 21.8 | -- | -- | -- |
| | | 3733 in³/cyl | Derate 25% | 6.0 | 0.30 | 0.50 | -- | 8160 | 4.1 | 24.0 | -- | -- | -- |
| | | 137.5 hp/cyl | | | | | | | | | | | |
| | | Ingersoll-Rand | | | | | | | | | | | |
| | | Model P:VGR-12 | | | | | | | | | | | |
| | | Ref. 16 | | | | | | | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

# TABLE C-45. EMISSIONS SOURCE TEST DATA -- ENGINE 45

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{Btu}{hp\text{-}hr}$ | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 45 | Gas | 133 bmep | Uncontrolled | 20.0 | 0.9 | 1.7 | -- | 7500 | -- | -- | -- | -- | -- |
| | | 1500 hp | Retard 5° | 15.0 | 0.9 | 1.7 | -- | 7540 | 0.5 | -- | -- | -- | -- |
| | | 550 rpm | Retard 10° | 10.0 | 0.9 | 1.7 | -- | 7760 | 3.5 | -- | -- | -- | -- |
| | | 4 cycle | Decrease manifold air temperature to 20° | 16.0 | 1.0 | 1.8 | -- | 7500 | 0 | -- | -- | -- | -- |
| | | Turbocharged | | | | | | | | | | | |
| | | Aftercooled 140°F | Decrease manifold air temperature to 20° | 12.5 | 1.4 | 2.0 | -- | 7425 | -1.0 | -- | -- | -- | -- |
| | | 12 cylinder | | | | | | | | | | | |
| | | 11.5" bore x 13" stroke | Derate 25% | 9.0 | 1.8 | 3.1 | -- | 7760 | 3.5 | -- | -- | -- | -- |
| | | 1350 in³/cyl | | | | | | | | | | | |
| | | 125 hp/cyl | | | | | | | | | | | |
| | | Ingersoll-Rand | | | | | | | | | | | |
| | | Model 12 SVS | | | | | | | | | | | |
| | | Ref. 16 | | | | | | | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-46. EMISSIONS SOURCE TEST DATA -- ENGINE 46

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 46 | Gas | 170 bmep | Uncontrolled | 16.5 | 1.2 | 2.8/<br>0.65 | -- | 6500 | -- | -- | -- | -- | -- |
|  |  | 6000 hp |  |  |  |  |  |  |  |  |  |  |  |
|  |  | 350 rpm | Derate 25% | 14.5 | 1.6 | 3.0 | -- | 6630 | 2.0 | -- | -- | -- | -- |
|  |  | 4 cycle | Retard 4° | 14.0 | 1.1 | 3.1 | -- | 6630 | 2.0 | -- | -- | -- | -- |
|  |  | Turbocharged | Retard 7° | 12.0 | 1.2 | 3.5 | -- | 6860 | 5.5 | -- | -- | -- | -- |
|  |  | Aftercooled 110° |  |  |  |  |  |  |  |  |  |  |  |
|  |  | 16 cylinders |  |  |  |  |  |  |  |  |  |  |  |
|  |  | 17" bore x 22" stroke |  |  |  |  |  |  |  |  |  |  |  |
|  |  | 4993 in³/cyl |  |  |  |  |  |  |  |  |  |  |  |
|  |  | 375 hp/cyl |  |  |  |  |  |  |  |  |  |  |  |
|  |  | Ingersoll-Rand |  |  |  |  |  |  |  |  |  |  |  |
|  |  | Model 616 KVR |  |  |  |  |  |  |  |  |  |  |  |
|  |  | Ref. 16 |  |  |  |  |  |  |  |  |  |  |  |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-47. EMISSIONS SOURCE TEST DATA -- ENGINE 47

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 47 | Diesel | 225 bmep<br>7313 hp<br>450 rpm<br>4 cycle<br>Turbocharged<br>Aftercooled 100°F<br>12 cylinder<br>17" bore x 21" stroke<br>4767 in$^3$/cyl<br>609 hp/cyl<br>DeLaval<br>Model DSRV-16<br>Ref. 17 | Uncontrolled | 9.7 | -- | 0.17 | -- | 6485 | -- | 29.5 | 54 | 71 | 29.96 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

# TABLE C-48. EMISSIONS SOURCE TEST DATA -- ENGINE 48

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 48 | Gas | 218 bmep<br>3200 hp<br>630 rpm<br>4 cycle<br>Turbocharged<br>Aftercooled 100°F<br>8 cylinders<br>14" bore x 15" stroke<br>2309 in³/cyl<br>400 hp/cyl<br>DeLaval<br>Model HVA-8<br>Ref. 17 | Uncontrolled | 19.0 | -- | 3.5 | -- | 6782 | -- | 26.3 | 55 | 76 | 29.84 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-49. EMISSIONS SOURCE TEST DATA -- ENGINE 49

T-886

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 49 | Gas | 182 bmep<br>2300 hp<br>630 rpm<br>4 cycle<br>Turbocharged<br>Aftercooled 100°F<br>8 cylinders<br>13" bore x 15" stroke<br>1991 in³/cyl<br>288 hp/cyl<br>DeLaval<br>Model HV-8<br>Ref. 17 | Uncontrolled | 18.2 | -- | 3.6 | -- | 7021 | -- | 27.3 | 51 | 68 | 29.99 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-50. EMISSIONS SOURCE TEST DATA -- ENGINE 50

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 50 | Dual fuel | 225 bmep<br>8658 hp<br>400 rpm<br>4 cycle<br>Turbocharged<br>Aftercooled 125°F<br>16 cylinders<br>17" bore x 21" stroke<br>4767 in³/cyl<br>541 hp/cyl<br>DeLaval<br>Model DGSRV-16<br>Ref. 17 | Uncontrolled | 4.4 | -- | 2.1 | -- | 6225 | -- | 27.9 | 71 | 82 | 29.80 |

T-656

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-51. EMISSIONS SOURCE TEST DATA -- ENGINE 51

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{Btu}{hp\text{-}hr}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 51 | Diesel | 225 bmep<br>4880 hp<br>450 rpm<br>4 cycle<br>Turbocharged<br>Aftercooled 125°F<br>8 cylinders<br>17" bore x 21" stroke<br>4767 in$^3$/cyl<br>610 hp/cyl<br>DeLaval<br>Model DSR-8<br>Ref. 17 | Uncontrolled | 8.5 | -- | 0.31 | -- | 6650 | -- | 29.4 | 63 | 65 | 29.74 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-52. EMISSIONS SOURCE TEST DATA -- ENGINE 52

T-686

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 52 | Diesel | 224 bmep<br>3631 hp<br>450 rpm<br>4 cycle<br>Turbocharged<br>Aftercooled 125°F<br>6 cylinders<br>17" bore x 21" stroke<br>4767 in³/cyl<br>605 hp/cyl<br>DeLaval<br>Model DSR-6<br>Ref. 17 | Uncontrolled | 9.4 | -- | 0.27 | -- | 6302 | -- | 31.2 | 63 | 66 | 29.74 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-53. EMISSIONS SOURCE TEST DATA -- ENGINE 53

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 53 | Gas | 187 bmep<br>5500 hp<br>630 rpm<br>4 cycle<br>Turbocharged<br>Aftercooled 100°F<br>16 cylinders<br>14" bore x 15" stroke<br>2309 in³/cyl<br>344 hp/cyl<br>DeLaval<br>Model HVA-16<br>Ref. 17 | Uncontrolled | 16.5 | -- | 3.2 | -- | 6714 | -- | 25.9 | 41 | 67 | 30.02 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-54. EMISSIONS SOURCE TEST DATA -- ENGINE 54

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 54 | Gas | 182 bmep<br>4000 hp<br>630 rpm<br>4 cycle<br>Turbocharged<br>Aftercooled 100°F<br>12 cylinder<br>14" bore x 15" stroke<br>2309 in$^3$/cyl<br>333 hp/cyl<br>DeLaval<br>Model HVA-12<br>Ref. 17 | Uncontrolled | 16.1 | -- | 2.5 | -- | 6851 | -- | 27.4 | 42 | 69 | 29.85 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-55. EMISSIONS SOURCE TEST DATA -- ENGINE 55

T-698

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp. °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 55 | Diesel | 240 bmep<br>1336 hp<br>1100 rpm<br>4 cycle<br>Turbocharged<br>Aftercooled<br>176°<br>6 cyl<br>9" bore x 10.5" stroke<br>668 in³/cyl<br>223 hy/cyl<br>White/Alco<br>6-25]F<br>Ref 18 | Uncontrolled | 7.3 | 1.8 | 0.3 | 12 | 6962 | -- | 29 | 45 | 90 | 29.29 |
| | | | Retard 2° | 7.2 | 2.0 | 0.2 | 14 | 7072 | 1.6 | 29 | 45 | 90 | 29.24 |
| | | | Manifold air temp. decrease by 76°F | 5.4 | 2.2 | 0.2 | 18 | 6980 | 0.3 | 31 | 45 | 80 | 29.29 |
| | | | Retard 2°, + Manifold Air Temp. dec. by 76°F | 5.4 | 1.9 | 0.2 | 19 | 7035 | 1.1 | 33 | 45 | 85 | 29.26 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-56. EMISSIONS SOURCE TEST DATA -- ENGINE 56

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 56 | Diesel | 214 bmep<br>1300 hp<br>900 rpm<br>4 cycle<br>Turbocharged<br>Aftercooled<br>8 cylinder<br>9" bore<br>10.5" stroke<br>668 in³/cyl<br>163 hp/cyl<br>White/Alco<br>Model 8-251F<br>Ref. 18 | Uncontrolled | 8.8 | 0.5 | 0.1 | -- | 6668 | -- | -- | 45 | 90 | -- |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-57. EMISSIONS SOURCE TEST DATA -- ENGINE 57

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 57 | Diesel | 240 bmep<br>2675 hp<br>1100 rpm<br>4 cycle<br>Turbocharged<br>Aftercooled<br>12 cylinder<br>9" bore x 10.5" stroke<br>668 in³/cyl<br>223 hp/cyl<br>White/Alco<br>Model 21-251F<br>Ref. 18 | Uncontrolled | 8.4 | 1.3 | 0.3 | 23 | 6943 | -- | 33 | 45 | 85 | 29.34 |

T-656

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-58. EMISSIONS SOURCE TEST DATA -- ENGINE 58

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 58 | Diesel | 230 bmep<br>2800 hp<br>1200 rpm<br>4 cycle<br>Turbocharged<br>Aftercooled<br>12 cylinder<br>9" bore x 10.5" stroke<br>668 in³/cyl<br>233 hp/cyl<br>White/Alco<br>Model 12-251F<br>Ref. 18 | Uncontrolled | 4.7 | 1.8 | 0.2 | 28 | 7328 | -- | 33 | 45 | 80 | 29.10 |
| | | | Injector hole dec. | 7.3 | 1.0 | 0.3 | 21 | 7181 | -2.0 | -- | 45 | 80 | 29.10 |

T-666

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-59. EMISSIONS SOURCE TEST DATA -- ENGINE 59

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 59 | Diesel | 262 bmep<br>3900 hp, intermittent rating<br>1100 rpm<br>4 cycle<br>Turbocharged<br>Aftercooled<br>16 cylinders<br>9" bore x 10.5 stroke<br>668 $in^3$/cyl<br>244 hp/cyl<br>White/Alco<br>Model 16-251F<br>Ref. 18 | Uncontrolled | 7.9 | 1.3 | 0.5 | -- | 6888 | -- | -- | 45 | 85 | -- |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-60. EMISSIONS SOURCE TEST DATA -- ENGINE 60

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 60 | Diesel | Large, high-speed, 4-cycle turbocharged engine. Greater than 350 in³/cyl | Uncontrolled | 8.3 | 1.6 | 0.3 | Bosch 1.4 | 6870 | -- | 28.6 | 35 | 69 | 29.25 |
| | | | Decrease A/F | 6.7 | 6.7 | 0.1 | Bosch 3.6 | 7511 | 9.3 | 21.1 | 35 | 69 | 29.25 |
| | | | Retard 4° | 5.8 | 1.9 | 0.2 | Bosch 2.2 | 7017 | 2.1 | 28.6 | 50 | 69 | 29.08 |
| | | | Retard 8° | 4.7 | 1.9 | 0.2 | Bosch 2.3 | 7383 | 7.5 | 28.6 | 47 | 71 | 29.09 |
| | | | Retard 12° | 4.0 | 1.5 | 0.3 | Bosch 2.3 | 7603 | 10.7 | 29.0 | 40 | 70 | 29.27 |
| | | | Retard 16° | 3.2 | 1.5 | 0.2 | Bosch 2.2 | 8152 | 18.7 | 27.6 | 32 | 73 | 29.04 |
| | | | Decrease manifold air temp. 10° (to 130°F) | 7.6 | 2.1 | 0.2 | Bosch 2.0 | 6998 | 1.9 | 28.6 | 49 | 75 | 29.17 |
| | | | Decrease manifold air temp.30° to 110°F | 7.5 | 1.5 | 0.2 | Bosch 1.7 | 6925 | 0.8 | 28.6 | 37 | 70 | 29.23 |
| | | | Retard 8° + A/F dec. | 4.2 | 4.5 | 0.2 | Bosch 3.8 | 8207 | 19.5 | 23.5 | 47 | 71 | 29.09 |
| | | | Dec. Manifold Air temp. 30° + A/F dec. | 6.7 | 3.3 | 0.2 | Bosch 2.6 | 7053 | 2.7 | 26 | 37 | 70 | 29.23 |
| | | | Retard 16° + A/F dec. | 2.8 | 3.8 | 0.1 | Bosch 3.7 | 8647 | 25.9 | 21.5 | 32 | 73 | 29.04 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

# TABLE C-61. EMISSIONS SOURCE TEST DATA -- ENGINE 61

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 61 | Dual fuel | 152.2 bmep | Uncontrolled | 7.3 | 2.0 | 1.5 | -- | 6570 | -- | 29.8 | 122 | 92 | 29.07 |
|  |  | 711 hp | Derate 25% | 5.9 | 3.0 | 3.9 | -- | 6970 | 6.1 | 30.4 | 164 | 92 | 28.81 |
|  |  | 900 rpm | Increase speed to 1000 rpm at constant power (137 bmep) | 5.1 | 2.7 | 4.4 | -- | 6960 | 5.9 | 30.6 | 99 | 92 | 28.83 |
|  |  | 4 cycle |  |  |  |  |  |  |  |  |  |  |  |
|  |  | Turbocharged |  |  |  |  |  |  |  |  |  |  |  |
|  |  | Aftercooled to 100°F |  |  |  |  |  |  |  |  |  |  |  |
|  |  | 6 cylinder |  |  |  |  |  |  |  |  |  |  |  |
|  |  | 9.125" bore x 10.5" stroke |  |  |  |  |  |  |  |  |  |  |  |
|  |  | 687 in³/cyl |  |  |  |  |  |  |  |  |  |  |  |
|  |  | 119 hp/cyl |  |  |  |  |  |  |  |  |  |  |  |
|  |  | White Superior |  |  |  |  |  |  |  |  |  |  |  |
|  |  | Model 40-GDSX-6 |  |  |  |  |  |  |  |  |  |  |  |
|  |  | Ref. 10 |  |  |  |  |  |  |  |  |  |  |  |

T-656

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[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-62. EMISSIONS SOURCE TEST DATA -- ENGINE 62

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 62 | Diesel | 152 bmep | Uncontrolled | 4.8 | 3.9 | 0.18 | -- | 7040 | -- | 36.3 | 110 | 92 | 28.66 |
| | | 711 hp | Derate 25% | 6.6 | 2.0 | 0.27 | -- | 6880 | -2.3 | 38.4 | 110 | 92 | 28.66 |
| | | 900 rpm | Increase speed to 1000 rpm at constant power (137 bmep) | 5.4 | 3.1 | 0.13 | -- | 7060 | +0.3 | 37.7 | 96 | 92 | 29.39 |
| | | 4 cycle | | | | | | | | | | | |
| | | Turbocharged | | | | | | | | | | | |
| | | Aftercooled to 100°F | | | | | | | | | | | |
| | | 6 cylinder | | | | | | | | | | | |
| | | 9.125" bore x 10.5" stroke | | | | | | | | | | | |
| | | 687 in³/cyl | | | | | | | | | | | |
| | | 119 hp/cyl | | | | | | | | | | | |
| | | White Superior | | | | | | | | | | | |
| | | 40 -x- 6 | | | | | | | | | | | |
| | | Ref. 10 | | | | | | | | | | | |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

T-656

## TABLE C-63. EMISSIONS SOURCE TEST DATA -- ENGINE 63

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 63 | Gas | Large, low-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in³/cyl | 110° Manifold Air Temperature (80° Ambient) |  |  |  |  |  |  |  |  |  |  |
|  |  |  | Uncontrolled | 16.5 | -- | -- | -- | 6820 |  | -- | 48 | 80 | -- |
|  |  |  | Lean A/F and retard 3° | 10.0 | -- | -- | -- | 7090 | 04.0 | -- | 46 | 80 | -- |
|  |  |  | Lean A/F, retard 4°, 30°F manifold temp. reduction | 4.3 | -- | -- | -- | 7430 | 8.9 | -- | 48 | 70 | -- |
|  |  |  | Reduced CR, lean A/F | 13.2 | -- | -- | -- | 8080 | 18.5 | -- | 62 | 80 | -- |
|  |  |  | Reduced CR, lean A/F, retard 5° | 8.5 | -- | -- | -- | 8360 | 22.6 | -- | 54 | 80 | -- |
|  |  |  | 130°F Manifold Air Temperature (80°F Ambient, Derated 8%) |  |  |  |  |  |  |  |  |  |  |
|  |  |  | Lean A/F, reduced CR | 14.7 | -- | -- | -- | 8180 | 19.9 | -- | 83 | 100 | -- |
|  |  |  | Lean A/F, retard 3° | 11.2 | -- | -- | -- | 8230 | 20.7 | -- | 83 | 100 | -- |
|  |  |  | Lean A/F, retard 3°, 50°F manifold temp. reduction | 5.3 | -- | -- | -- | 8370 | 22.7 | -- | 83 | 70 | -- |
|  |  |  | Lean A/F ratio | 17.0 | -- | -- | -- | 7010 | 2.8 | -- | 48 | 100 | -- |
|  |  |  | Lean A/F ratio and 3° ignition retard | 10.3 | -- | -- | -- | 7200 | 5.6 | -- | 48 | 100 | -- |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-64. EMISSIONS SOURCE TEST DATA -- ENGINE 64

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 64 | Gas | Large, low-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in³/cyl | 110°F Manifold Air Temperature (80°F Ambient) | | | | | | | | | | |
| | | | Uncontrolled | 11.7 | -- | -- | -- | 6760 | -- | -- | 106 | 80 | -- |
| | | | Retard 4° | 10.7 | -- | -- | -- | 7077 | 4.7 | -- | 106 | 80 | -- |
| | | | 30°F manifold reduction | 8.1 | -- | -- | -- | 6900 | 2.1 | -- | -- | 80 | -- |
| | | | Retard 4°, 30°F manifold reduction | 7.0 | -- | -- | -- | 7150 | 5.8 | -- | 62 | 80 | -- |
| | | | Derate 10% | 9.4 | -- | -- | -- | 6900 | 2.1 | -- | 62 | 80 | -- |
| | | | 130°F Manifold Air Temperature (80°F Ambient Derated 8%) | | | | | | | | | | |
| | | | Uncontrolled | 12.9 | -- | -- | -- | 6837 | 1.1 | -- | 63 | 100 | -- |
| | | | Derate 20% | 13.1 | -- | -- | -- | 6820 | 0.9 | -- | 64 | 100 | -- |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-65. EMISSIONS SOURE TEST DATA -- ENGINE 65

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | NOx | CO | HC[a] | | | | | | | |
| 65 | Gas | Large, low-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in³/cyl | 110°F Manifold Air Temperature (80°F Ambient) | | | | | | | | | | |
| | | | Uncontrolled | 17.6 | -- | -- | -- | 7265 | | -- | 58 | 80 | -- |
| | | | Retard 3° | 8.9 | -- | -- | -- | 7510 | 3.4 | -- | 62 | 80 | -- |
| | | | Derate 20% | 10.2 | -- | -- | -- | 7580 | 4.3 | -- | 54 | 80 | -- |
| | | | 130°F Manifold Air Temperature (100°F Ambient Derated 8%) | | | | | | | | | | |
| | | | Uncontrolled | 16.0 | -- | -- | -- | 7410 | 2.0 | -- | 57 | 100 | -- |
| | | | Retard 2° | 8.2 | -- | -- | -- | 7530 | 3.6 | -- | -- | 100 | -- |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

# TABLE C-66. EMISSIONS SOURCE TEST DATA -- ENGINE 66

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 66 | Gas | Large, low-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in³/cyl | 110°F Manifold Air Temperature (80°F Ambient) | | | | | | | | | | |
| | | | Uncontrolled | 14.0 | — | — | — | 6780 | | — | 64 | 80 | -- |
| | | | Lean A/F | 10.5 | — | — | — | 6820 | 0.6 | — | 68 | 80 | -- |
| | | | Lean A/F, retard 3° | 8.3 | — | — | — | 7100 | 4.7 | — | 68 | 80 | -- |
| | | | Lean A/F, 30°F manifold reduction | 7.3 | — | — | — | 6895 | 1.7 | — | 84 | 80 | -- |
| | | | Reduced Compression Ratio | | | | | | | | | | |
| | | | Lean A/F | 12.2 | — | — | — | 7560 | 11.5 | -- | 79 | 80 | — |
| | | | Lean A/F, retard 4° | 8.5 | — | — | — | 7910 | 16.7 | -- | 75 | 80 | -- |
| | | | Lean A/F, 30° manifold reduction | 11.1 | — | — | — | 7775 | 14.7 | -- | 84 | 80 | -- |
| | | | 130°F Manifold Air Temperature (100°F Ambient, Derate 8%) | | | | | | | | | | |
| | | | Uncontrolled | 13.7 | — | — | — | 6850 | 1.0 | -- | 70 | 100 | -- |
| | | | Lean A/F | 13.2 | — | — | — | 7550 | 11.4 | -- | 74 | 100 | -- |
| | | | Lean A/F, retard 4° | 9.4 | — | — | — | 7740 | 14.2 | — | 79 | 100 | -- |
| | | | Lean A/F, 20°F mainfold reduction | 10.6 | — | — | — | 7830 | 15.5 | -- | 90 | 80 | -- |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

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## TABLE C-67. EMISSIONS SOURCE TEST DATA -- ENGINE 67

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 67 | Gas | Large, low-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in$^3$/cyl | 110°F Manifold Air Temperature (80°F Ambient) |  |  |  |  |  |  |  |  |  |  |
|  |  |  | Lean A/F | 8.7 | -- | -- | -- | 6670 |  | -- | 51 | 80 | -- |
|  |  |  | 130°F Manifold Air Temperature (100°F Ambient Derated 8%) |  |  |  |  |  |  |  |  |  |  |
|  |  |  | Uncontrolled | 12.7 | -- | -- | -- | 6660 |  | -- | 51 | 100 | -- |
|  |  |  | Lean A/F | 10.5 | -- | -- | -- | 6803 | 2.1 | -- | 73 | 100 | -- |
|  |  |  | Lean A/F, 30°F manifold reduction | 9.8 | -- | -- | -- | 6650 | -0.2 | -- | 56 | 100 | -- |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-68. EMISSIONS SOURE TEST DATA -- ENGINE 68

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 68 | Diesel | Large, low-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in$^3$/cyl | Uncontrolled | 15.5 | — | — | — | 6618 | 100.0 | — | 50 | 75 | — |
| | | | 30°F manifold air temperature reduction (to 100°F) | 14.6 | — | — | — | 6540 | 98.8 | — | 48 | 75 | — |
| | | | Derate 8% | 15.8 | — | — | — | 6676 | 100.9 | — | 44 | 75 | — |
| | | | Derate 8%, 30°F manifold air temperature reduction | 14.9 | — | — | — | 6540 | 98.8 | — | 47 | 75 | — |

T-656

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-69. EMISSIONS SOURCE TEST DATA -- ENGINE 69

T-666

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 69 | Gas | Large, medium-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in³/cyl | Uncontrolled | 10.3 | 1.33 | 2.48/<br>2.62 | -- | 6942 | -- | 32.0 | 36 | 70 | 28.59 |
| | | | A/F increase 4.4% | 8.0 | 1.86 | 2.72/<br>2.49 | -- | 7301 | 5.2 | 33.4 | 102 | 70 | 29.01 |
| | | | Derate 25%, A/F increase | 4.4 | 2.76 | 4.66 | -- | 7942 | 14.4 | 34.6 | 102 | 70 | 29.01 |

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[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-70. EMISSIONS SOURCE TEST DATA -- ENGINE 70

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 70 | Gas | Large, medium-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in³/cyl | Uncontrolled | 8.6 | 1.66 | 3.1/<br>3.72 | -- | 6845 | -- | 37.4 | 15 | 70 | 28.74 |
| | | | EGR 6.5% | 5.8 | 1.46 | 3.05/<br>3.64 | -- | 6847 | 0.0 | 36.1 | 13 | 70 | 28.60 |
| | | | EGR 7.3%<br>Derate 17% | 4.4 | 2.51 | 4.1/<br>4.7 | -- | 7210 | 5.3 | 35.3 | 17 | 70 | 28.77 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-71. EMISSIONS SOURCE TEST DATA -- ENGINE 71

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 71 | Gas | Large medium-speed, 2-cycle, blower scavenged engine. Greater than 350 in$^3$/cyl | Uncontrolled | 11.5 | 1.4 | 2.8 | -- | 7400 | -- | 33.4 | -- | 65 | 29.12 |
| | | | Derate 20% (constant speed) | 4.6 | 2.2 | 3.0 | -- | 7700 | 4.1 | 33.4 | -- | 65 | 29.12 |
| | | | Derate 20% by decreased speed (const. torque) | 7.0 | 1.8 | 3.6 | -- | 6900 | -6.2 | 37.0 | -- | 65 | 29.12 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-72. EMISSIONS SOURCE TEST DATA -- ENGINE 72

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{Btu}{hp-hr}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 72 | Diesel | Large, medium-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in$^3$/cyl | Uncontrolled | 12.6 | 0.71 | 0.25 | Bosch 0.4 | 6508 | -- | 36.5 | 20 | 70 | 29.15 |
| | | | Derate 25% | 12.7 | 0.72 | 0.23 | Bosch 0.4 | 6601 | 1.4 | 39.5 | 22 | 70 | 29.14 |
| | | | Air-to-fuel increase, manifold air temperature increase | 9.1 | 0.65 | 0.29 | Bosch 0.3 | 6714 | 3.2 | 41.8 | 4 | 70 | 29.66 |

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[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-73. EMISSIONS SOURCE TEST DATA -- ENGINE 73

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 73 | Diesel | Large, medium-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in³/cyl. | Uncontrolled | 9.3 | 0.57 | 0.22 | -- | 6809 | -- | 41.7 | 21 | 70 | 29.06 |
|  |  |  | CR,IEGR | 9.6 | 0.62 | 0.18 | -- | 6666 | -2.1 | 41.5 | 19 | 70 | 29.34 |

T-656

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-74. EMISSIONS SOURCE TEST DATA -- ENGINE 74

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 74 | Diesel | Large, medium-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in$^3$/cyl | Uncontrolled | 8.5 | 0.65 | 0.22 | Bosch 0.7 | 6741 | -- | -- | 67 | 70 | 29.18 |
| | | | Retard 3° | 6.7 | 0.84 | 0.25 | Bosch 0.6 | 6907 | 2.5 | 00 | 62 | 70 | 29.17 |

T-656

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-75. EMISSIONS SOURCE TEST DATA -- ENGINE 75

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 75 | Diesel | Large, low-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in³/cyl | Uncontrolled | 6.2 | 3.9 | 0.4 | Bosch 1.2 | 7035 | -- | 30.8 | -- | 80 | 29.18 |
| | | | A/F change | 9.4 | 1.4 | 0.4 | Bosch 0.6 | 6668 | -- | 32.8 | 52 | 80 | 29.48 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

T-656

## TABLE C-76. EMISSIONS SOURE TEST DATA -- ENGINE 76

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 76 | Dual fuel | Large, low-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in³/cyl | Uncontrolled | 10.6 | 2.89 | 1.25 | Bosch 0.5 | 6208 | -- | 35.0 | -- | 80 | 29.18 |
| | | | Retard & A/F change | 5.4 | 2.12 | 3.06 | Bosch 0.2 | 6264 | 0.9 | 34.6 | 64 | 80 | 29.58 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-77. EMISSIONS SOURE TEST DATA -- ENGINE 77

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] $\frac{Btu}{hp\text{-}hr}$ | % Change in BSFC | A/F | Humidity gr H2O/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 77 | Diesel | Large, medium-speed, 2-cycle turbocharged, aftercooled engine. Greater than 350 $in^3$/cyl | Uncontrolled | 11.4 | 0.91 | 0.22 | Bosch 0.5 | 6504 | -- | 40.5 | 37 | 70 | 29.07 |
| | | | Retard 5°, change A/F | 7.9 | 1.08 | 0.20 | Bosch 0.7 | 6797 | 4.5 | 40.8 | 24 | 90 | 29.50 |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

# TABLE C-78. EMISSIONS SOURCE TEST DATA -- ENGINE 78

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 78 | Dual fuel | Large, medium-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in³/cyl. | Uncontrolled | 7.4 | 1.20 | 1.00 | Bosch 0.05 | 6297 | -- | 38.2 | 35 | 70 | 29.07 |
| | | | Retard 2.5°, A/F increase | 3.25 | 1.52 | 2.17 | Bosch 0.05 | 6495 | 3.1 | 42.0 | 34 | 70 | 29.15 |

T-656

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-79. EMISSIONS SOURCE TEST DATA -- ENGINE 79

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp., °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 79 | Gas | Medium-bore, high speed, naturally aspirated engine. | Uncontrolled | 18.0 | 2.2 | 1.9 | -- | -- | -- | -- | -- | -- | -- |

T-456

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-80. EMISSIONS SOURCE TEST DATA -- ENGINE 80

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp. °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 80 | Gas | Medium-bore, high speed, naturally aspirated engine. | Uncontrolled | 17.2 | 8.0 | 1.7 | -- | -- | -- | -- | -- | -- | -- |

T-656

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## TABLE C-81. EMISSIONS SOURCE TEST DATA -- ENGINE 81

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr | | | Visible Emission, Percent Opacity | BSFC,[b] Btu/hp-hr | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp. °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | $NO_x$ | CO | $HC_T$[a] | | | | | | | |
| 81 | Gas | Medium-bore, high speed, turbocharged, aftercooled engine. | Uncontrolled | 16.9 | 1.4 | 0.7 | -- | -- | -- | -- | -- | -- | -- |

T-886

[a] Total hydrocarbons

[b] Brake specific fuel consumption

# TABLE C-82. EMISSIONS SOURCE TEST DATA -- ENGINE 82

T-656

| Engine Code No. | Fuel | Description | Control | Emissions, g/hp-hr |  |  | Visible Emission, Percent Opacity | BSFC,[b] $\frac{\text{Btu}}{\text{hp-hr}}$ | % Change in BSFC | A/F | Humidity gr $H_2O$/lb air | Temp. °F | Barometric Pressure, inch Hg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  |  | $NO_x$ | CO | $HC_T$[a] |  |  |  |  |  |  |  |
| 82 | Gas | Medium-bore, high speed, turbocharged, aftercooled engine. | Uncontrolled | 14.4 | 1.1 | 0.8 | -- | -- | -- | -- | -- | -- | -- |

[a] Total hydrocarbons

[b] Brake specific fuel consumption

## C.2 REVIEW OF AMBIENT CORRECTION FACTORS FOR APPLICATION TO LARGE-BORE ENGINE $NO_x$ DATA

Ambient correction factors have been developed for automobile- and truck-size gasoline spark ignition (SI) and compression ignition (CI) engines(23). The following sections will discuss the application of the existing SI factors to natural gas engines (C.2.1) and the existing CI factors to large-bore diesel and dual-fuel engines (C.2.2). In addition, gas turbine ambient correction factors will be examined for application to large-bore engine emissions (C.2.3). As these sections will illustrate, no satisfactory ambient temperature correction factor has been developed for any size of SI internal combustion engines, and only one study has considered ambient temperature corrections for CI engines. Therefore, Section C.2.4 will discuss an analytical approach to correct emissions for variations in ambient temperature.

### C.2.1 Ambient Correction Factors Developed for SI Engines

A survey of the literature showed that correction factors for gasoline-fueled engines have been developed only for ambient humidity variations(24). One of the studies also evaluated the effect of ambient temperature and barometric pressure variations on exhaust emissions, but found that engine-to-engine variations were too great to generalize a correction factor for either temperature or pressure(25). These studies will be briefly discussed in the following paragraphs, and then the ambient humidity correction factors will be compared and evaluated for application to the large-bore natural gas-fueled engines in this study.

The Automobile Manufacturer's Association contracted with the Ethyl Corporation to conduct a study on the effect of ambient air humidity, temperature, and pressure on heavy-duty (HD) gasoline engines(26). The results of this study for the effect of humidity later became the correction factor designated in the Federal Register for HD gasoline engines(27).

This correction factor was derived from emission measurements conducted on seven engines in accordance with the Gasoline-Fueled, Heavy-Duty Engine procedures in the 1970 Federal Register. The test engines were installed in an engine dynamometer test cell where humidity, temperature, and pressure were varied. The engines tested were gasoline-fueled, spark-ignited, carbureted, heavy-duty truck engines ranging in size from 38 to 75 CID per cylinder. The compression ratios varied from 7 to 9.4, and air-to-fuel ratios varied from 14.4 to 15.7. The objective of the study was to develop factors to adjust composite mass emissions to a standard condition of 75 grains $H_2O$/lb dry air, 90°F inlet air temperature, and a barometric pressure of 29.92 inches of mercury.

Figure C-1(28) illustrates the effect of humidity on both A/F ratio and $NO_x$ emissions (Federal nine-mode composite cycle). As this figure illustrates, a reasonably good correlation was established between changes in ambient humidity and $NO_x$ emissions. Note also that A/F ratios are essentially constant or decrease slightly with increasing inlet air humidity. The composite cycle ambient correction factor was of the form

$$K = 0.634 + 0.00654(H) - 0.0000222(H)^2 \qquad \text{(C-1)}$$

where $NO_x$ corrected = (K) $NO_x$ observed

H = specific humidity, grains $H_2O$/lb dry air

Figures C-2(29) and C-3(30) illustrate the effect of temperature and pressure on A/F and $NO_x$ emissions for these same engines. Obviously, engine-to-engine variations were too great to generalize a correction factor for either temperature or barometric pressure.

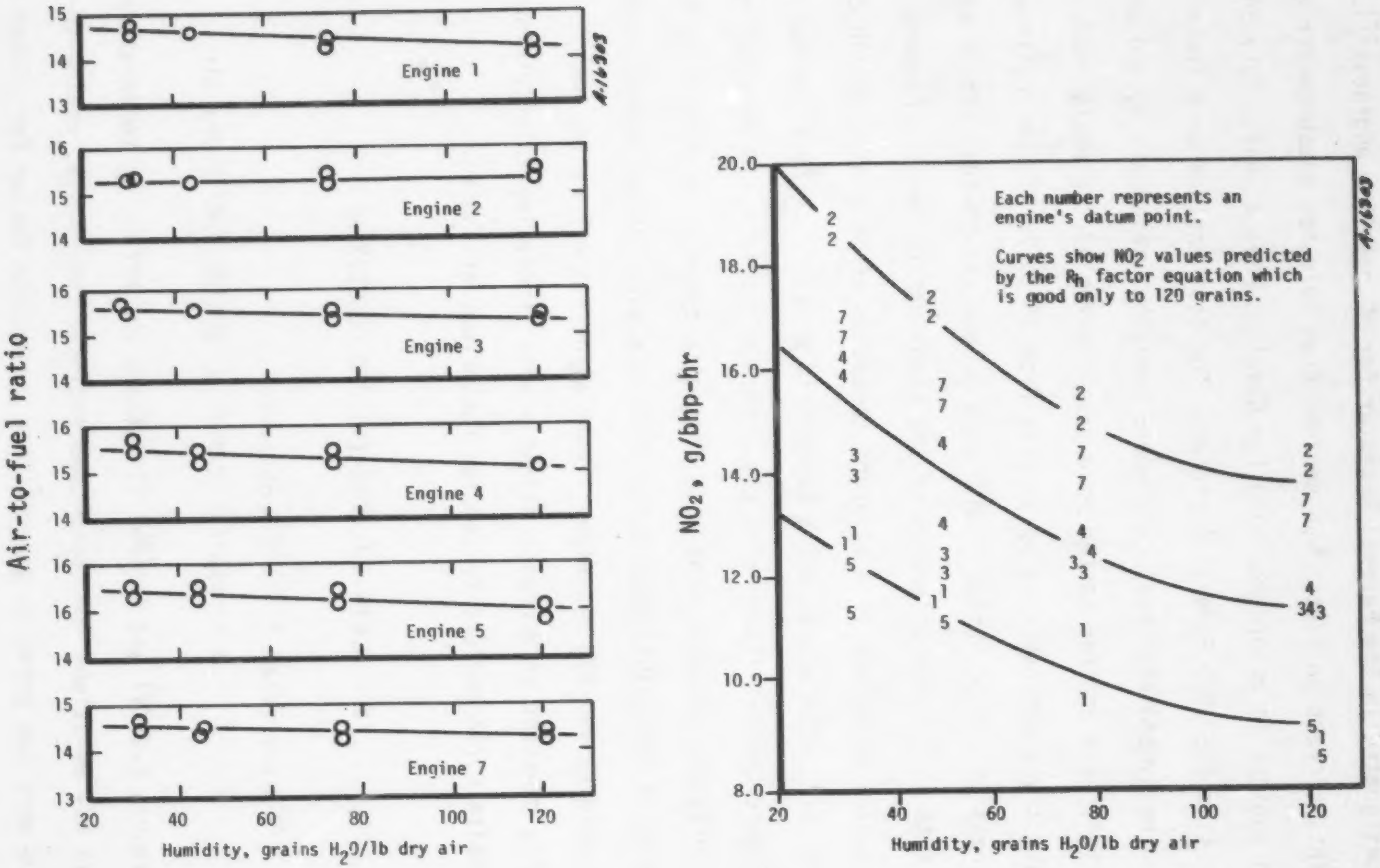

Figure C-1. Effect of humidity on A/F ratio and $NO_x$ emissions for HD gasoline engines (28).

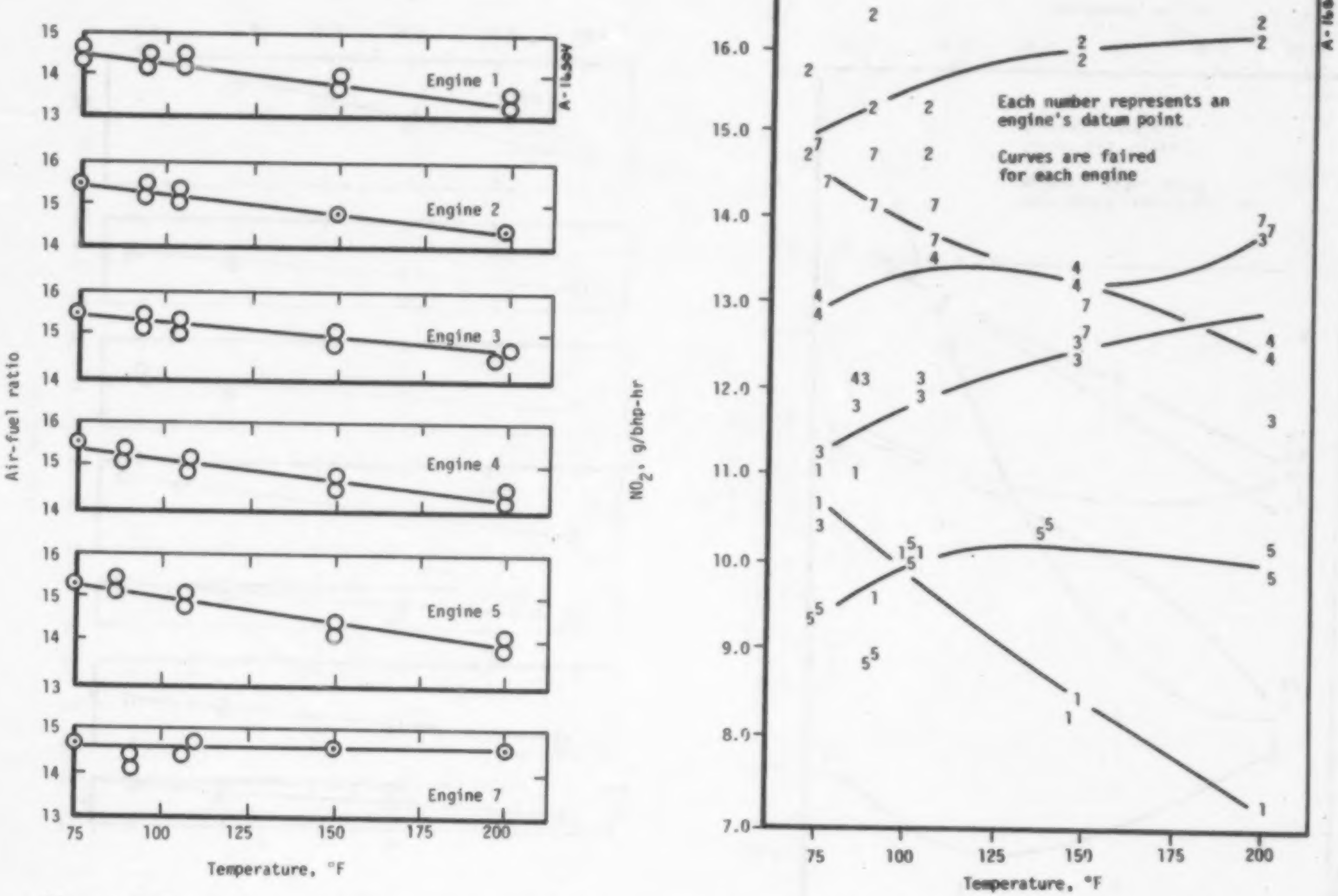

Figure C-2. Effect of temperature on A/F ratio and $NO_x$ emissions for HD gasoline engines (29).

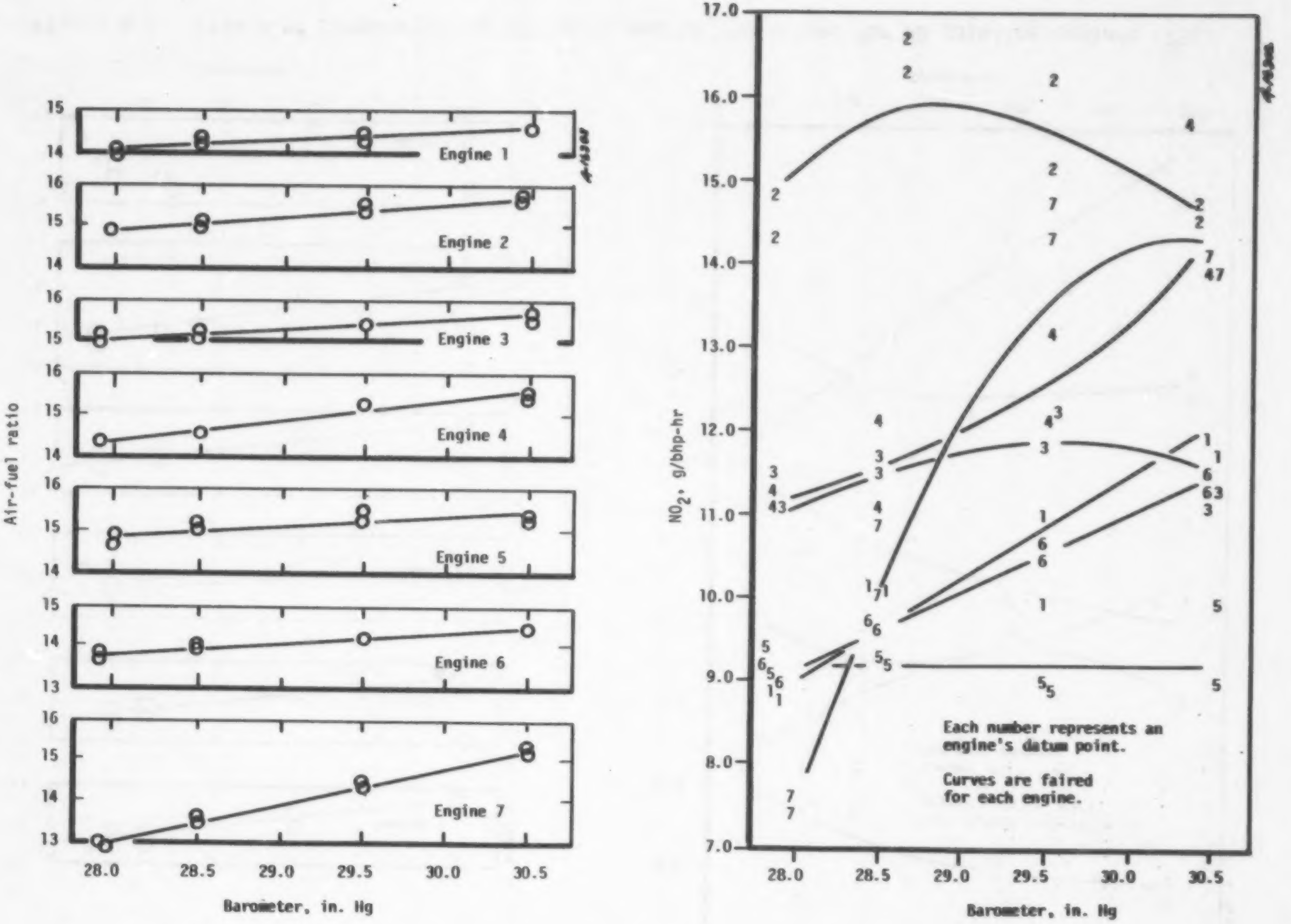

Figure C-3. Effect of barometer pressure on A/F ratio and $NO_x$ emissions for HD gasoline engines (30).

Both physical reasoning and these results indicate that changes in ambient temperature and pressure affect the A/F ratio more than do changes in humidity. Moreover, it is well known that the effect of A/F ratio variations on $NO_x$ emissions depends on the engine operating point on the $NO_x$ vs. A/F curve (see Figure C-4)(31). For example, a decrease in A/F ratio for an engine operating at A in Figure C-4 would result in a decrease in $NO_x$ emissions, whereas a decrease in A/F ratio for an engine at C would cause an increase in $NO_x$ emissions. The author concluded, therefore, that engines operating at different A/F ratios with different metering characteristics should exhibit varying (possibly contradictory) effects on exhaust emissions for similar changes in inlet air conditions.

The author also indicated that changes in inlet air conditions could affect fuel distribution to the engine, and this in turn affects $NO_x$ emissions. This effect is illustrated in Figure C-5(32). In addition, changes in inlet air conditions which change the engine A/F ratio also change the minimum spark advance for best torque (mbt). Figure C-6(33) illustrates how $NO_x$ emissions vary with A/F for the mbt setting and other settings retarded from mbt. Since some of the engines in that study had no vacuum advances, they experienced, in effect, retarded mbt settings as their A/F ratio changed. Thus, their emissions would respond differently to A/F changes than engines with vacuum advances. Therefore, the development of a general correction factor for ambient temperature and pressure was not possible, although variations in these parameters affected emissions significantly.

In two other studies, ambient humidity corrections were developed from LD gasoline vehicles(34,35). In the first study, correction factor was derived from emission tests at 76°F and at four ambient humidities on a fleet of eight passenger cars operated on a chassis dynamometer set to simulate

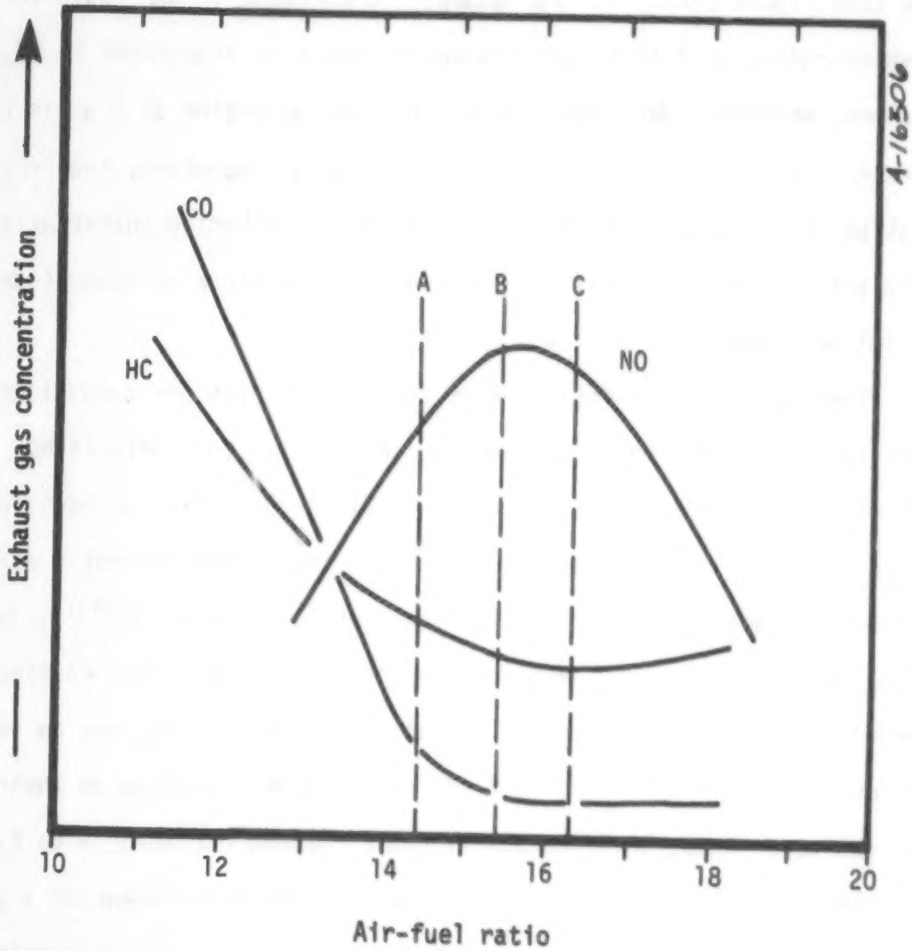

Figure C-4. Exhaust gas concentration versus A/F (31).

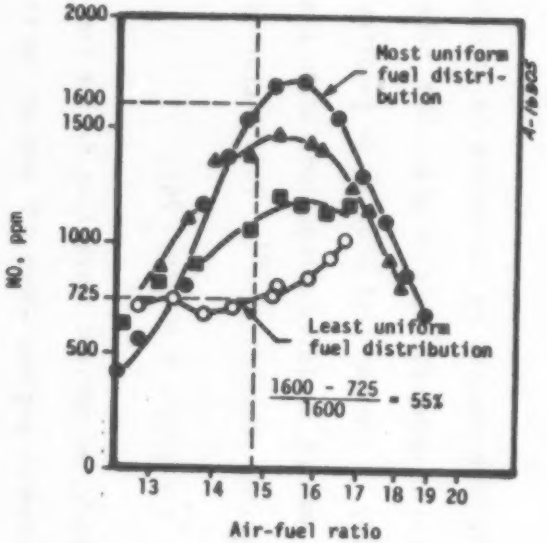

Figure C-5. Effect of fuel distribution on NO emissions (32).

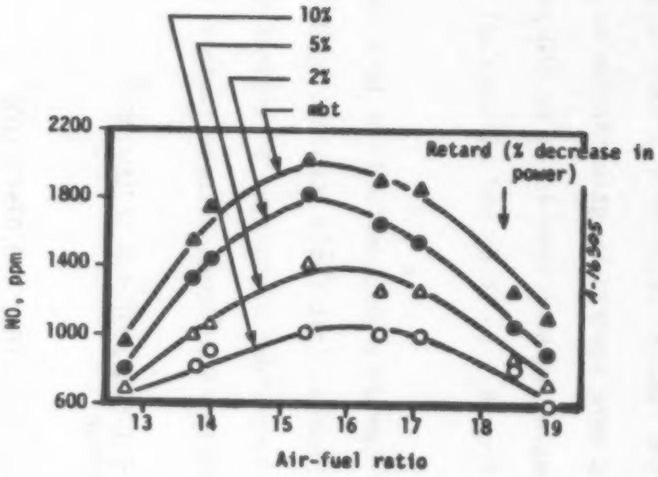

Figure C-6. Effect of ignition timing and A/F on NO exhaust emissions, 1600 rpm, 23 bhp road at mbt (33).

derived from emission tests at 76°F and at four ambient humidities on a fleet of eight passenger cars operated on a chassis dynamometer set to simulate seven different road loads of the Federal seven-mode composite cycle for LD vehicles. These passenger cars were powered by gasoline-fueled, spark-ignited, carbureted engines ranging in size from 42 to 59 CID per cylinder. The compression ratios varied from 8.5 to 10.5, and air-to-fuel ratios from 14.6 to 16.4.

Corrections for ambient humidity were derived for both composite cycle and constant load. The Federal Test Cycle constant load factor is considered more applicable to large-bore carbureted SI engines which typically operate at a constant load (nearly rated load). These factors are

$$K = 0.796 + 0.175(H/100) + 0.129(H/100)^2 \quad \text{composite factor} \tag{C-2a}$$

$$K = 0.844 + 0.151(H/100) + 0.075(H/100)^2 \quad \text{constant load, 50 mph} \tag{C-2b}$$

The results of this study were adopted by California to correct emissions for ambient humidity for gasoline-powered vehicles under 6000 pounds(36).

In the second study, a similar test program on gasoline vehicles was conducted to develop an ambient humidity correction factor that was adopted by EPA to correct LD gasoline vehciles(37,38). This factor is

$$K = 1/(1 - 0.0047(H - 75)) \tag{C-3}$$

In yet another study, an ambient correction factor for humidity was developed for typical research engines where A/F and mbt settings were held constant(39). The author of this study also noted the effect of A/F ratio on $NO_x$ emissions as illustrated in Figure C-7(40). Based on this figure,

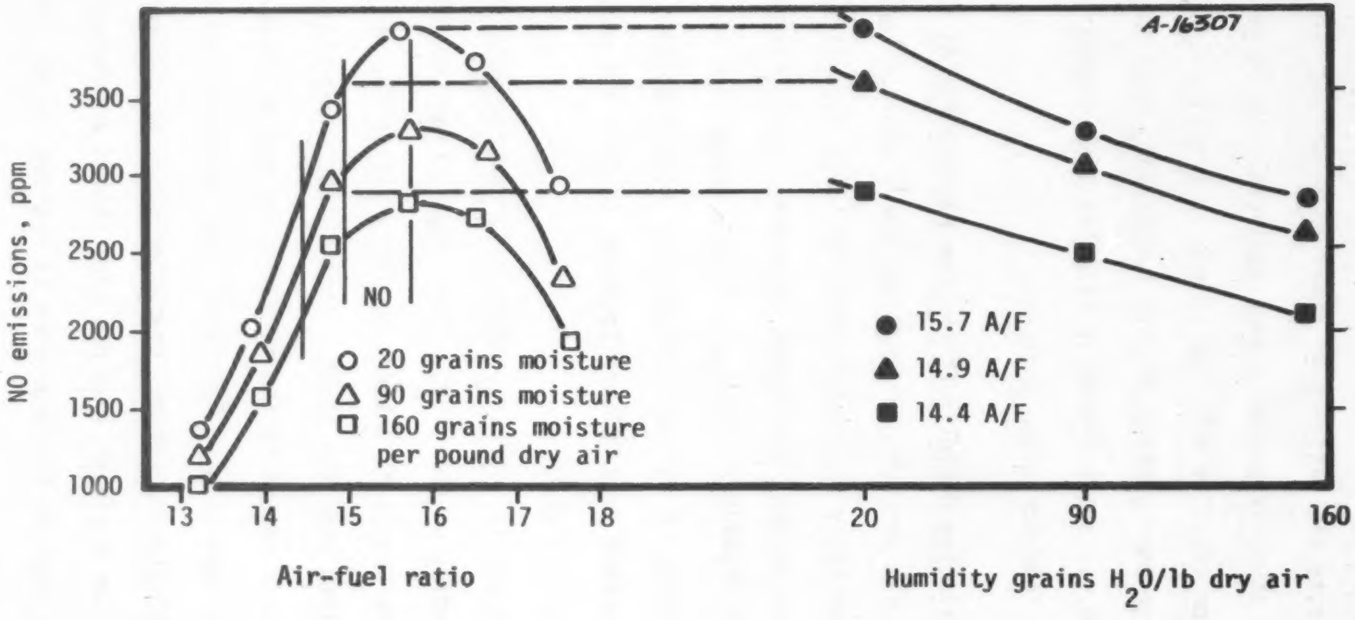

Figure C-7. Effect of A/F ratio and ambient humidity on NO emissions of a LD gasoline engine (40).

Krause(41) reasoned that composite correction factors would vary from constant load factors since $NO_x$ emissions vs. humidity curves changed with A/F ratio. Despite the validity of these observations, the correction factor derived in this study is probably less applicable to large-bore engines than those derived for gasoline vehicles, since in practice A/F ratios and spark settings of large-bore engines are not necessarily held constant as they were in this study (e.g., spark timing is fixed as load changes for some engines, and varies with load for other engines).

## Comparison of Existing Humidity Correction Factors for SI Engines

Based on the preceding discussion, three ambient humidity correction factors are potentially applicable to large-bore, natural gas fueled engines, particularly four-stroke, carbureted versions. These factors are summarized in Table C-83(42,43,44,45). Figure C-8 is a comparison of the three factors over a typical range of ambient humidities. Note that only one of the factors plotted is a constant load factor (Equation (C-2b); the other three are based on composite test cycles.

As Figure C-8 illustrates, there is a considerable difference in correction depending on the study. The results from the EPA/Scott study show the greatest sensitivity to ambient humidity variation, while the results for HD gasoline engines show the least. This is not unexpected in view of the previous discussion regarding the variation in engine responses to changes in inlet conditions depending on A/F ratio, fuel metering and distribution, and ignition (distribution operation) characteristics of different engines. The correction based on constant load has been chosen as the most suitable correction to be applied to large-bore SI engines, since these engines are

# TABLE C-83. AMBIENT HUMIDITY CORRECTION FACTORS FOR SI ENGINES

| Equation No. | Correction Factor |
|---|---|
| (1) | $K = 0.634 + 0.00654(H) - 0.0000222(H)^2$ Composite Factor (9 Mode Federal HD Gasoline Test Cycle) (42) |
| (2a) | $K = 0.796 + 0.175(H/100) + 0.129(H/100^2)$ Composite Factor (Federal Test Cycle, LD Gasoline Vehicles) (43) |
| (2b) | $K = 0.844 + 0.151(H/100) + 0.075(H/100)^2$ 50 mph, Constant Load (44) |
| (3) | $K = 1/(1 - 0.0047(H-75))$ Composite Factor (Federal Test Cycle, LD Gasoline Vehicles) (45) |

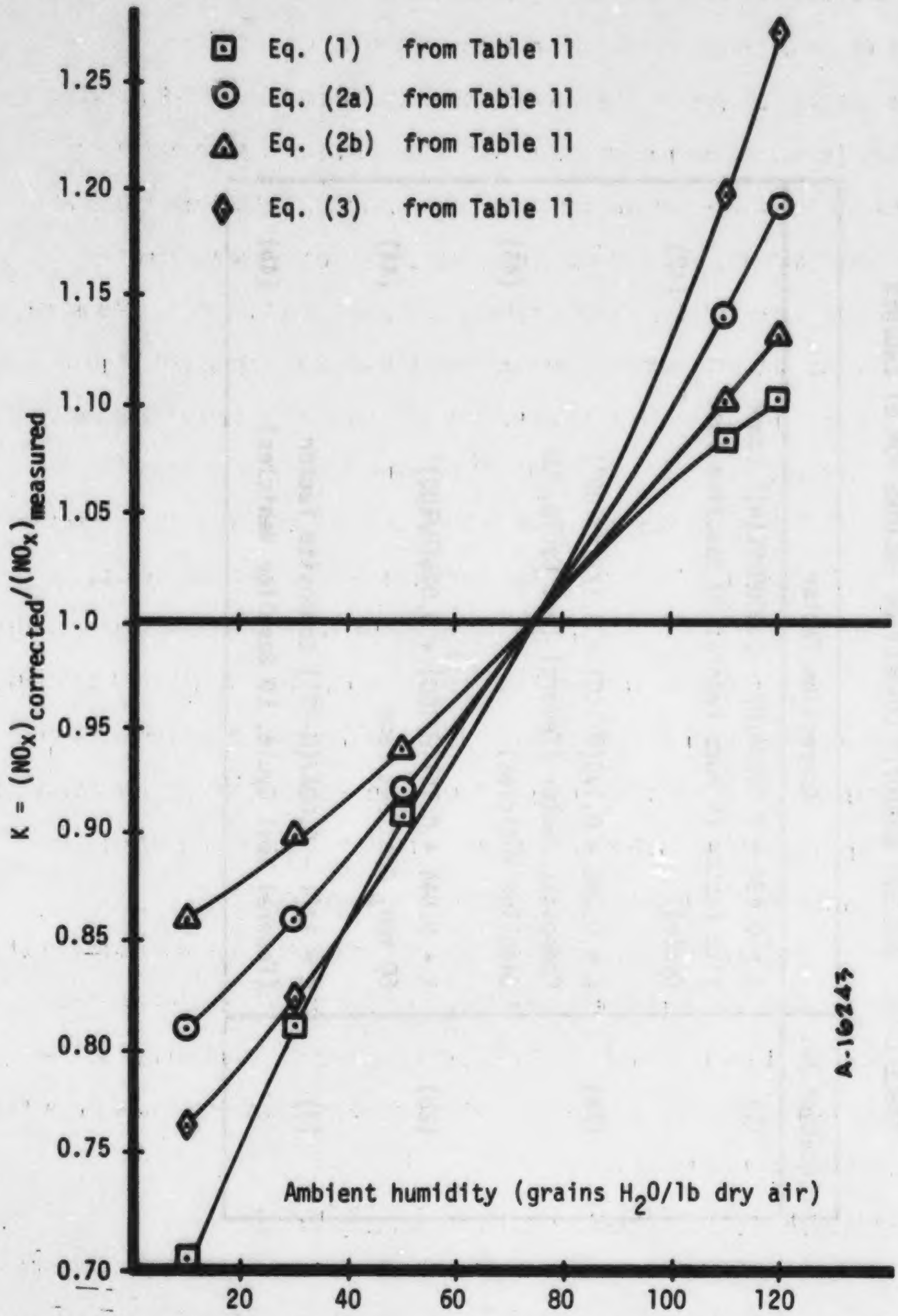

Figure C-8. Comparison of SI ambient humidity correction factors.

typically operated at constant load. Thus, this correction factor is most applicable to carbureted (4-NA) natural gas engines.

Application of any of the correction factors in Figure C-8 to other engine types (e.g., turbocharged units) is questionable due to major differences in inlet air intake systems. For example, data from the draft SSEIS indicate that $NO_x$ reductions due to water induction are directly proportional to water-to-fuel (W/F) rates, up to W/F ratios of 1. Moreover, W/F rates due to ambient humidity are a function of A/F ratio, as Figure C-9 illustrates. Based on this figure, the W/F rate of a carbureted natural gas engine (trapped A/F $\approx$ 17) is about 30 percent lower for a specific humidity of 100, than the W/F ratio of a turbocharged engine with a trapped A/F ratio of 25 (trapped A/F ratios for turbocharged SI engines typically range from 20 to 25)(46,47). Note, also, that the curves of constant humidity diverge with increasing A/F ratio. Therefore, it can be anticipated that $NO_x$ emissions of engines with different A/F ratios will respond differently to identical changes in ambient humidity. Thus, the application of the constant load humidity correction factor (based on carbureted gasoline engines) to other than carbureted, large-bore SI engines is questionable.

Similar conclusions can also be reasoned regarding the application of ambient temperature and pressure correction factors to $NO_x$ emissions from engines whose air intake as well as fuel systems differ substantially. As discussed earlier, however, no factors have been developed for SI engines for either temperature or pressure.

### C.2.2 Ambient Correction Factors Developed for CI Engines

A survey of the literature established that two sources have reported ambient correction factors for truck-size diesel engines(48,49). A study by Krause, et al., was sponsored by the Automobile Manufacturer's Association

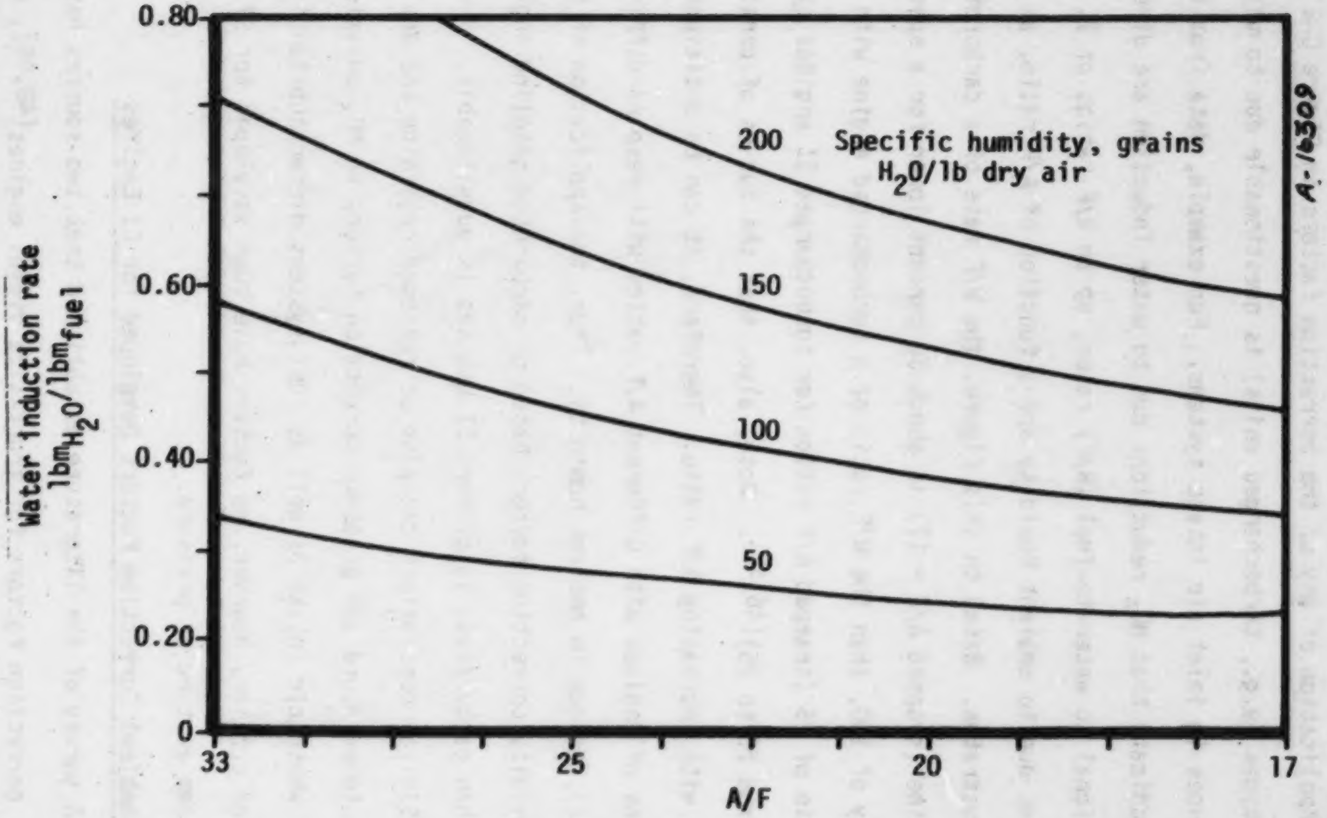

Figure C-9. Water-to-fuel range versus ratio air-to-fuel.

and the Engine Manufacturer's Association and resulted in correction factors for both temperature and humidity. These corrections were later adopted by the EPA for HD diesel engines(50). The other source reported corrections for humidity only.

In Krause's study, a correction factor was derived for six different engines run over a 13-mode cycle. Emission procedures used were similar to "California Procedures for Diesel Engines in 1973 and Subsequent Model Year Vehicles Over 6001 Pounds Gross Vehicle Weight." The heavy-duty truck engines tested were diesel-fueled, compression-ignition engines ranging in size from 69 to 155 CID. The compression ratios varied between 14.9 and 18.7. Inlet air conditions were controlled to one of 19 combinations of humidity and temperature. The humidity conditions ranged from 35 to 125 grains $H_2O$/lb dry air, and the temperature was varied between 70°F and 115°F. Barometric pressure was controlled to 28.00 ± 0.2 inch Hg at the air cleaner inlet.

The engines in the study were of the following types:

- Four-stroke turbocharged, direct injection (4-TC)
- Four-stroke turbocharged, prechamber (4-TC, PC)
- Four-stroke naturally aspirated, MAN chamber (4-NA, MAN)
- Four-stroke naturally aspirated, (4-NA)
- Two-stroke blower scavenged (2-BS)
- Four-stroke turbocharged, aftercooled (4-TC, AC)

Unlike turbocharged large-bore engines, which are nearly always aftercooled, only one of the turbocharged engines in this study was aftercooled. That engine as well as the 4-NA and the 2-BS units were similar in design to large-bore engines. A correction factor was developed for all of the engines as well as for individual units. The factor was of the form

$$K = 1/\left(A(H - 75) + B(T - 85)\right)$$

where NO corrected = $(K)_x$ (NO observed) and A and B are related to A/F ratio (load). The reference ambient conditions are 75 grains $H_2O$/lb dry air and 85°F. Figure C-10(51) is a plot of the coefficients A and B for each engine as a function of load (or A/F ratio).

Average values of A and B for all engines (as function of A/F) were also determined. This result does not appear wholly justified given the variations in response to ambient humidity and temperature exhibited by the different engine types depicted in Figure C-10. Note that the 4-TC, AC unit is significantly less sensitive to ambient temperature variations over the load range than the other designs. Therefore, separate ambient correction factors for 2-BS, 4-NA, and 4-TC, AC units from this study were used on the corresponding large-bore designs instead of just one average value of all engines. The coefficients A and B were determined for rated load conditions using average rated load A/F ratios that were reported for the large-bore engines.

The second source(52) which examined humidity effects reported correction factor based on experimental tests of a 2-BS and a 4-NA engine.

Ambient humidity corrections from both of these sources (assuming inlet temperature is held constant, hence, the B term of Krause's factor drops out) are illustrated in Figure C-11(53,54). The SI (gasoline) factors discussed in Section C.2.1 are also plotted for comparison. Note that there is little variation in correction factors for different diesel engine types with the exception of the 2-BS unit from the CRC report. Moreover, the diesel-fueled (CI) engines appear less sensitive to ambient humidity changes

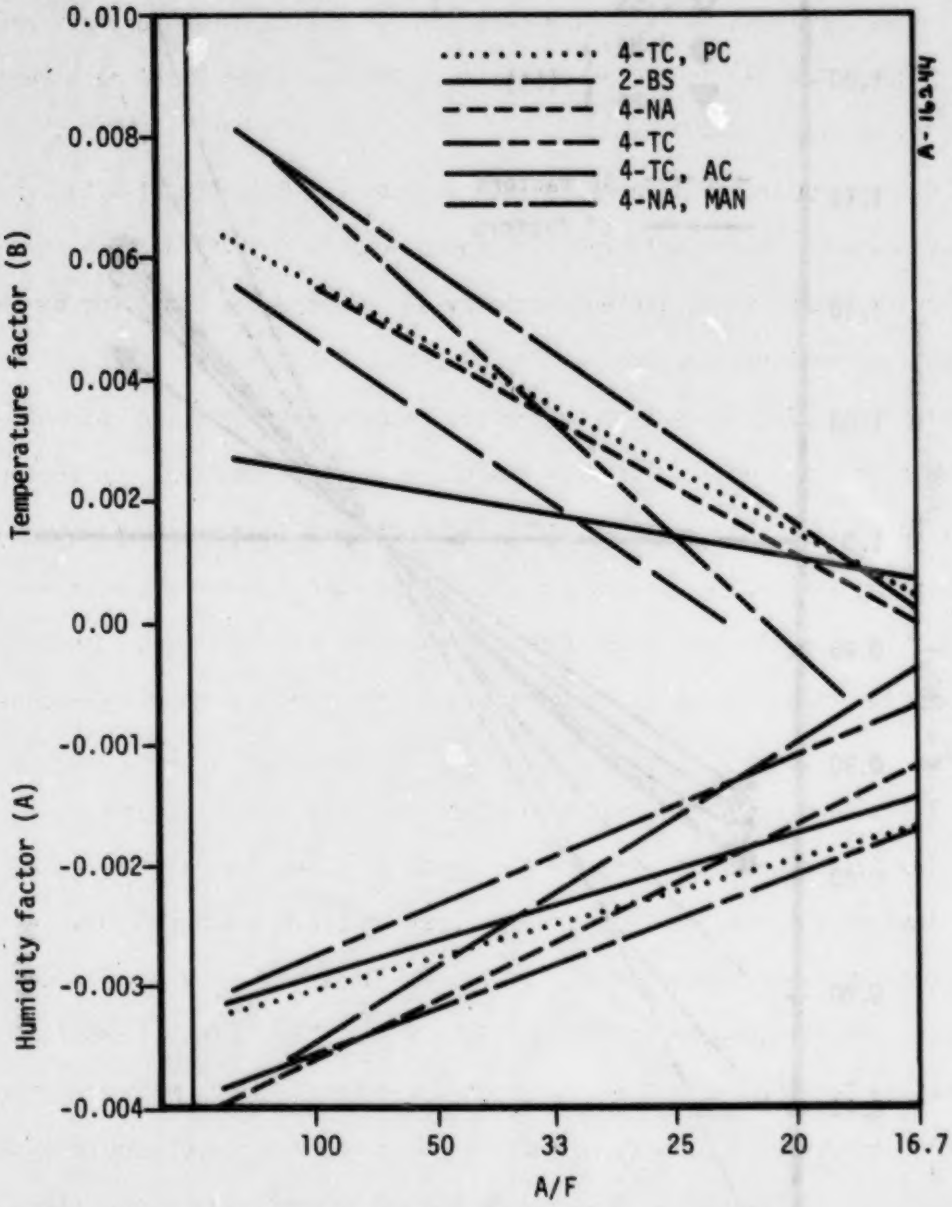

Figure C-10. Effect of humidity and temperature on NO emission for different CI engines (51).

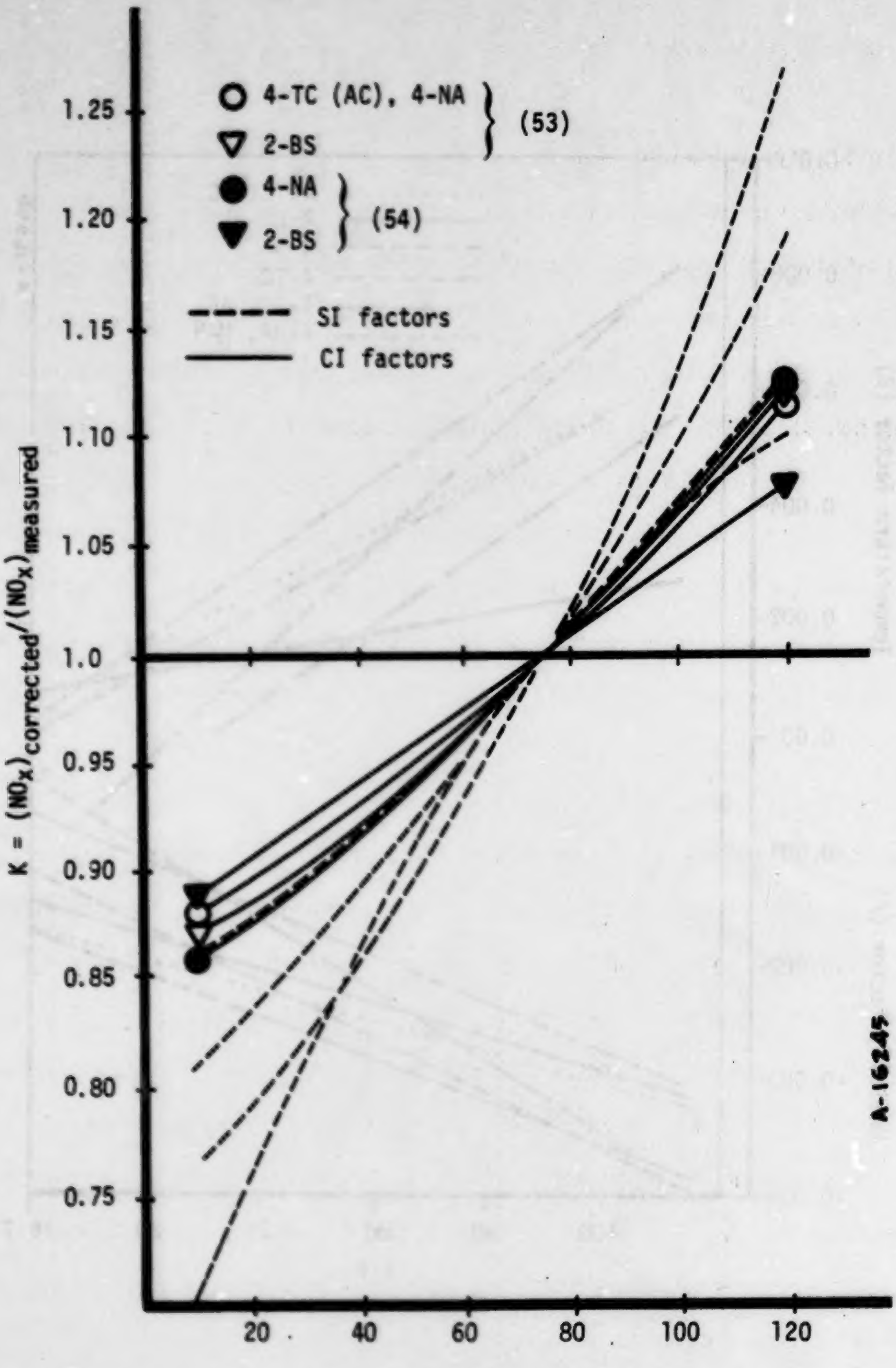

Figure C-11. Comparison of CI and SI ambient humidity correction factors.

than SI units, particularly for humidities less than 75 grains $H_2O$/lb dry air.

Initially, one might expect the diesel units to be more responsive to inlet humidity variations, since these units operate at higher A/F ratios (approximately 25 to 40), and therefore, induct more ambient water than SI engines which operate more nearly at stoichiometric ratios (≈15 to 17, depending on the fuel). Diesel engines, however, have a greater thermal inertia than SI engines due to their higher trapped A/F ratios (as well as a different combustion process). Apparently the higher thermal inertia in the diesel units more than offsets their higher effective water induction rate; thus, their $NO_x$ emissions are less sensitive to changes in ambient humidity than SI engines. This explanation is corroborated by experiments which have demonstrated that water injection, as a control technique, produces significantly greater $NO_x$ reductions in SI units than it does in CI (diesel) units (see Section 4.4.7 of the draft SSEIS).

The Krause study also investigated the effect of ambient temperature on $NO_x$ emissions. Figure C-12(55) presents the correction factors that were derived for engine types similar to those in the present study (humidity is assumed constant, hence, the A term of Krause's factor drops out). The reference temperature is taken as 85°F. This figure shows that the naturally aspirated and blower-scavenged engines $NO_x$ emissions are more sensitive to inlet air temperature changes than the aftercooled design. Since Krause's study systematically examined the effect of both inlet temperature and humidity for a number of CI engine types, his factors were selected for application to similar large-bore engine types.

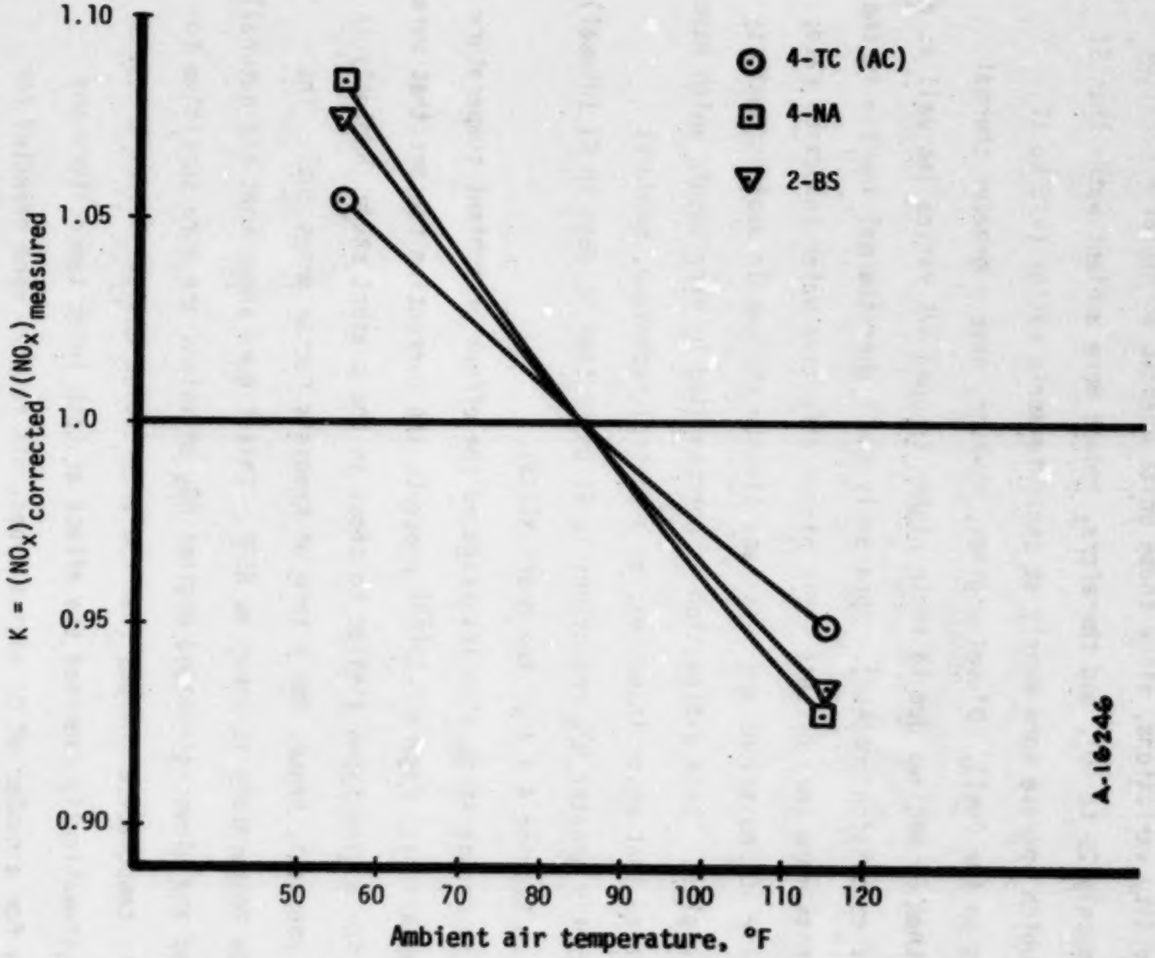

Figure C-12. Correction factors for temperature for CI engines (55).

### C.2.3 Potential Application of Gas Turbine Ambient Correction Factors to Reciprocating IC Engines

Since no ambient correction factors have been developed specifically for large-bore engines, all existing correction factors for internal combustion engines were examined including those for gas turbines. There is a considerable amount of technical literature on gas turbine ambient correction factors as a result of the promulgation of emission standards for aircraft engines and the proposed emission standards for stationary gas turbines(56-61). This section describes these various factors and their potential application to the large-bore engine data in the present study.

Gas turbine combustion characteristics have some common features with those in diesel engines. Both the gas turbine primary combustion zone and the diesel combustion chamber can be characterized as a well-stirred reactor, and both use similar distillate fuels or natural gas. On the other hand, the gas turbine combustion is a steady-state, constant pressure process, whereas the diesel is an unsteady, variable pressure combustion process. Nevertheless, the similarities of the two systems warrant an investigation of existing gas turbines correction factors.

Many investigators have developed ambient humidity correction factors based on a model that relates $NO_x$ formation parameters of temperature, pressure, equivalence ratio and residence time (using the kinetic rate equations for $NO_x$ formation)(62). Humidity enters this model through its effect on reaction (flame) temperature. Most researchers have shown that the effect of humidity on $NO_x$ formation takes the form

$$NO_x \text{ corrected}/NO_x \text{ observed} = \exp\left[K\left(H_{\text{observed}} - H_{\text{reference}}\right)\right]$$

where $H_{ref}$ = specific humidity at reference (standard) conditions

K = empirical constant that ranges from 14 to 30, generally taken as 19(63)

Figure C-13(64,65) is a plot of this correction factor and shows that it agrees reasonably well with the HD diesel (4-TC, AC) factor discussed in Section C.2.2 (the gas turbine factor was adjusted to a reference humidity of 75 grains $H_2O$/lb dry air to correspond with the reference humidity of the HD factor).

Other sources have shown similar agreement of humidity effects on $NO_x$ emissions of diesel engines and gas turbines as illustrated in Figure C-14(66). The gas turbine humidity correction factor was derived from empirical data based on water injection as a means of $NO_x$ control in gas turbines. The ambient humidity was converted into an effective water-to-fuel ratio by multiplying the ambient humidity loading by the near stoichiometric A/F ratio existing at combustion. Then the empirically derived water injection correction factor of Ambrose(67) was used to calculate the percentage reduction in $NO_x$. It is reasoned that the overall A/F ratio is inappropriate since much of the water vapor in the inlet air never reaches the primary combustion zone because it is vented for engine cooling or enters downstream as dilution or wall cooling air. Based on this result, one can conclude that changes in humidity appear to affect $NO_x$ formation in gas turbines in much the same way as diesel (IC) engines. However, it should be remembered that a number of adequate humidity correction factors have been developed for IC engines; therefore, this gas turbine result is of limited value, but serves to reinforce the more global application of these corrections.

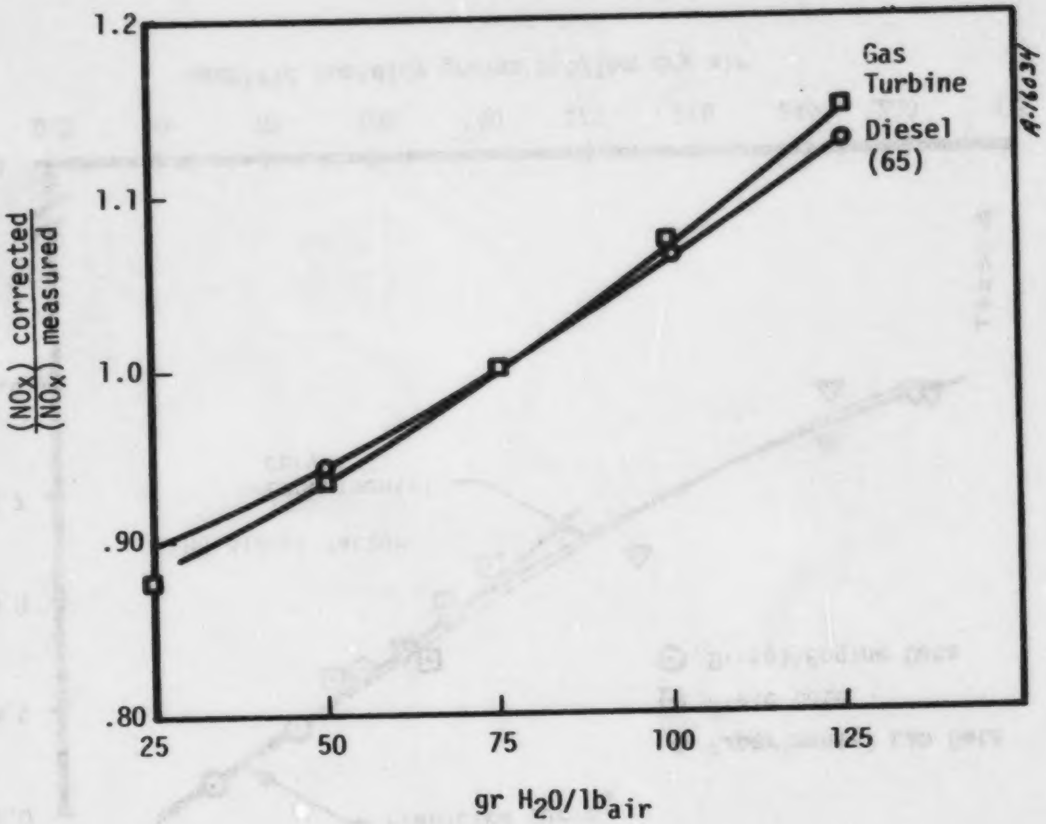

Figure C-13. Gas turbine humidity correction factor (64).

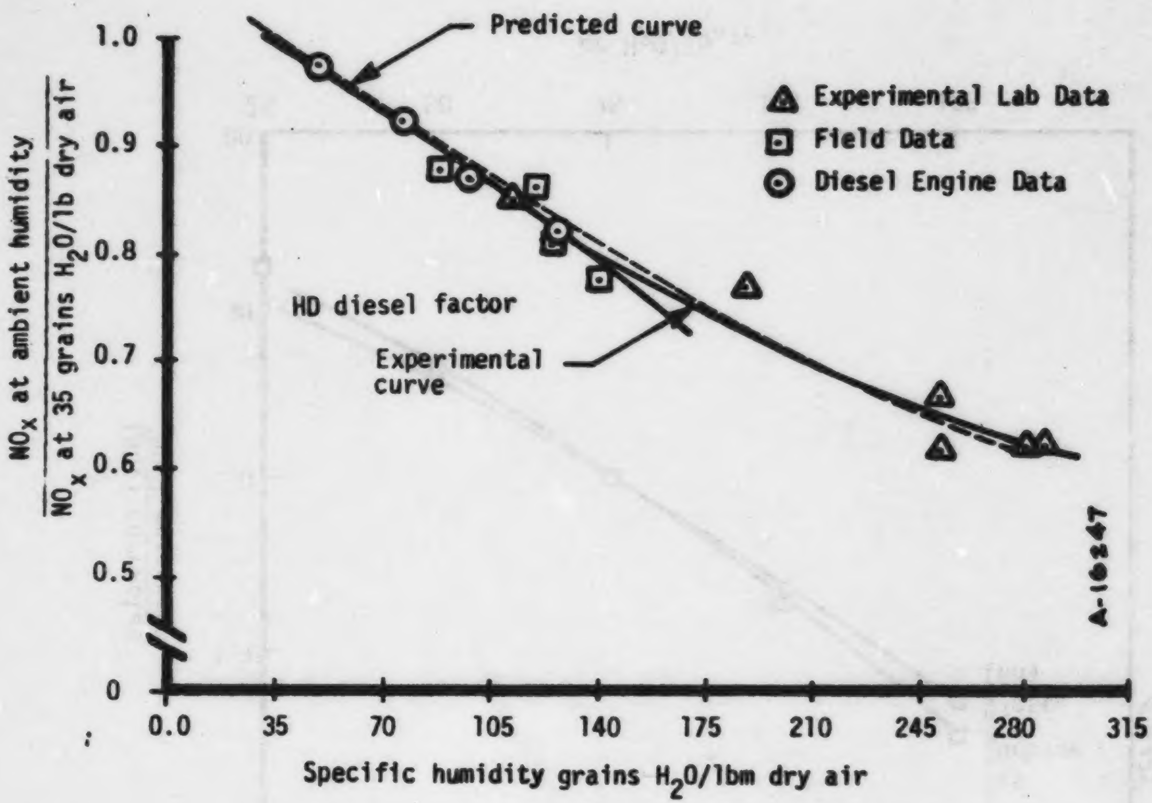

Figure C-14. Observed ambient humidity influence on $NO_x$ production(66).

Gas turbine temperature corrections were also examined for application to reciprocating engine data. Figure C-15(68,69) is a comparison of the HD diesel (4-TC, AC) correction factor with various gas turbine factors. Obviously, there is little agreement with the exception of one factor. Presumably these differences arise from differences in inlet air compressor and combustor design. The gas turbine factors, however, have some flexibility in that they are related to the combustor inlet parameters of pressure and temperature. Therefore, an attempt was made to relate changes in $NO_x$ formation of reciprocating engines to changes in ambient temperature by estimating the cylinder temperature before combustion in a typical reciprocating engine and using the compressor pressure ratio and the assumption of isentropic compression to calculate temperature. This calculated temperature was then used with the gas turbine factors. Figure C-16(70,71) presents the results of this approach for several different forms of gas turbine factors. Again, these results do not correlate well with the HD diesel factor, probably due to the empirical nature of the gas turbine equations and the large differences in air intake systems between engines and turbines.

On the basis of this brief review, gas turbine ambient correction factors do not make a useful contribution as potential ambient correction factors for reciprocating engines.

### C.2.4 An Analytical Approach for the Ambient Temperature Correction of $NO_x$ Emissions

Since no temperature correction factor has been reported in the literature (see Section C.2.1) for SI engines, and no systematic emissions data exist (either SI or CI) from which to base a temperature correction for large-bore engines in general, an attempt was made to develop a

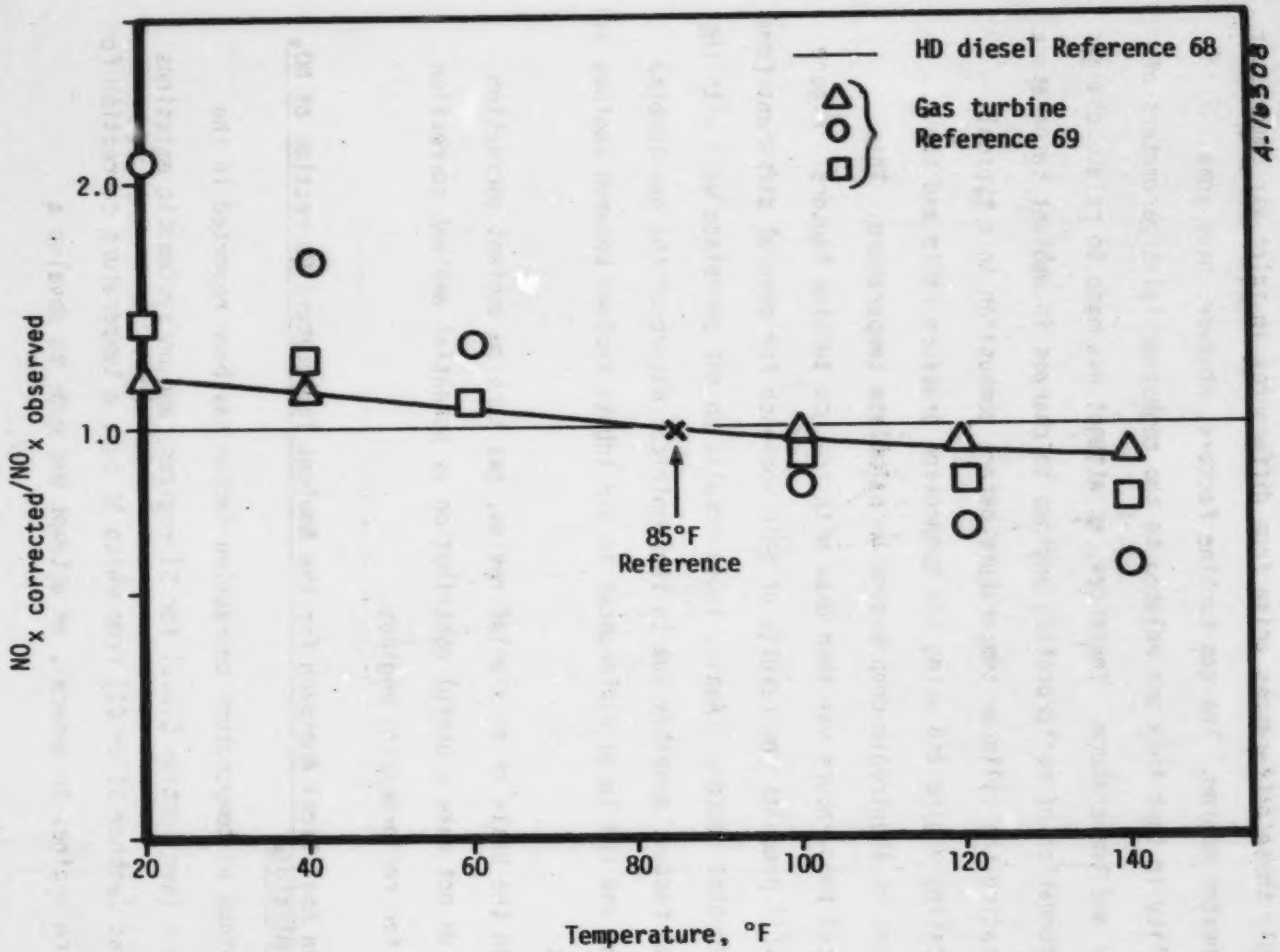

Figure C-15. Comparison of existing gas turbine ambient temperature correction with HD diesel temperature correction.

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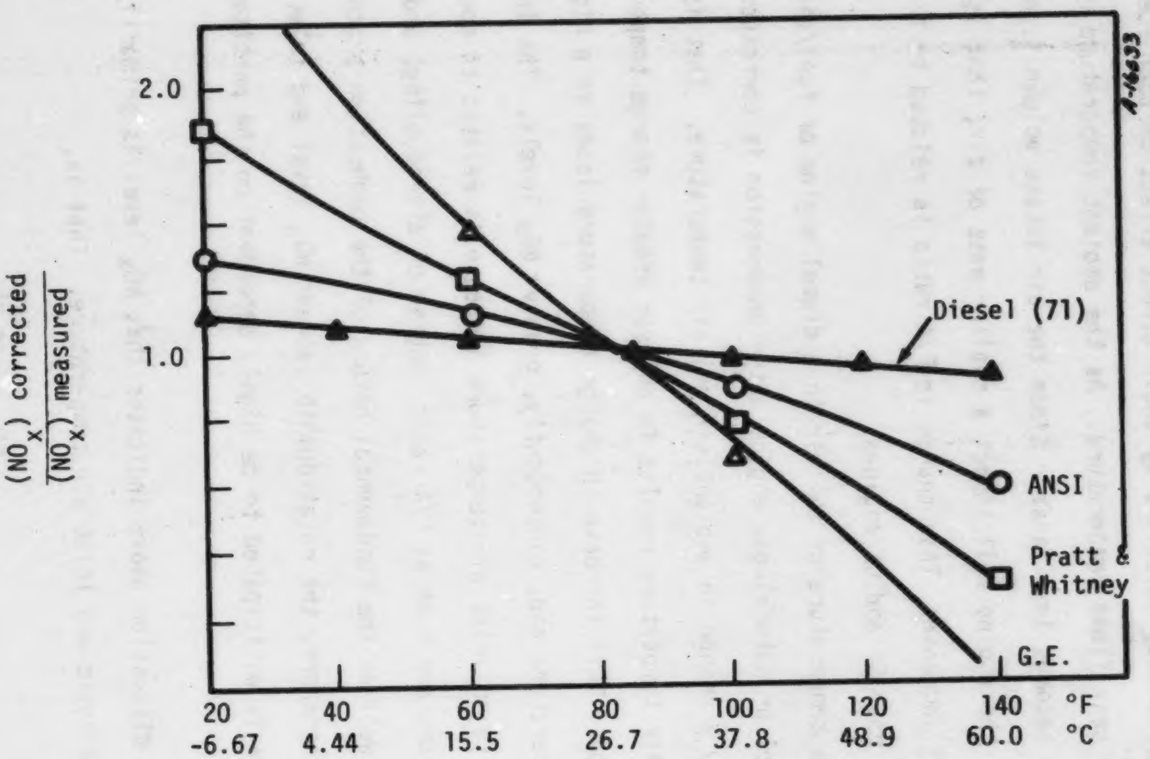

Figure C-16. Gas turbine temperature correction factors (70).

semianalytical approach for relating changes in ambient temperature to changes in $NO_x$ level from large-bore engines. This approach was used to estimate how changes in ambient temperature affect $NO_x$ levels.

The analytical approach is based on the fact that ambient temperature changes affect $NO_x$ emissions by their direct effect on both fuel/air (F/A) ratio and peak flame temperature. As the ambient temperature rises, the inlet air becomes less dense. Since the air intake volume is essentially constant, the engine will inject a smaller mass of air; that is, the F/A ratio will increase. This change in F/A ratio is related to a change in $NO_x$ level for both SI and CI engines.

The temperature of the air in a diesel engine or fuel/air mixture in a dual-fuel or natural-gas engine after compression is correspondingly changed by a change in ambient (inlet) air temperature. That is, an increase in inlet air temperture results in an even greater charge temperature after compression. This increase in charge temperature leads to a higher peak flame temperature and, consequently, greater $NO_x$ levels. The increase in $NO_x$ level due to an inlet air temperature increase is related to specific engine design parameters such as F/A ratio, degree of aftercooling, and compression ratio, as well as the fundamental nature of the combustion process (i.e., CI or SI). Therefore, the relationship between $NO_x$ level and inlet air temperature is anticipated to be highly dependent on the particular engine design.

The discussion above indicates that $NO_x$ level is primarily a function of both F/A ratio and inlet air temperature. That is,

$$NO_x = g(f,T)$$

where $f \equiv F/A$, fuel/air ratio

$T \equiv T_a$, ambient inlet temperature

$N \equiv NO_x$, oxides of nitrogen

The subscript R denotes a reference ambient condition

Since we are interested in predicting changes in $NO_x$ level rather than absolute $NO_x$ level we may mathematically represent a change in $NO_x$ to a change in ambient temperature as

$$dN = (\partial N/\partial f)_T \, df + (\partial N/\partial T)_f \, dT$$

or (C-4)

$$dN/dT = \partial N/\partial f \; df/dT + \partial N/\partial T$$

where the derivatives on the right side of Equation (C-4) remain to be evaluated. The first partial derivative, $\partial N/\partial f$, represents the change in $NO_x$ emissions due to a change in F/A at constant ambient temperature, while the second partial, $\partial N/\partial T$, represents the change in $NO_x$ due to temperature variations at constant F/A. This mathematical formulation can be portrayed graphically as shown in Figure C-17. The diagram shows $NO_x$ vs. fuel-to-air ratio, the downsloping curves, and load vs. fuel-to-air ratio, the upsloping curves, for a typical turbocharged diesel engine. The basic problem is to find the change in $NO_x$ at constant load due to some ambient temperature change. For example, assume the ambient temperature, T, is greater than the reference temperature, $T_R$. Starting at the uncorrected fuel-to-air ratio, one locates Point A on the $NO_x$ production curve and Point B on the load curve. Then moving over, at constant load, to the reference temperature load curve, one locates Point C and, hence, the reference fuel-to-air ratio. Now,

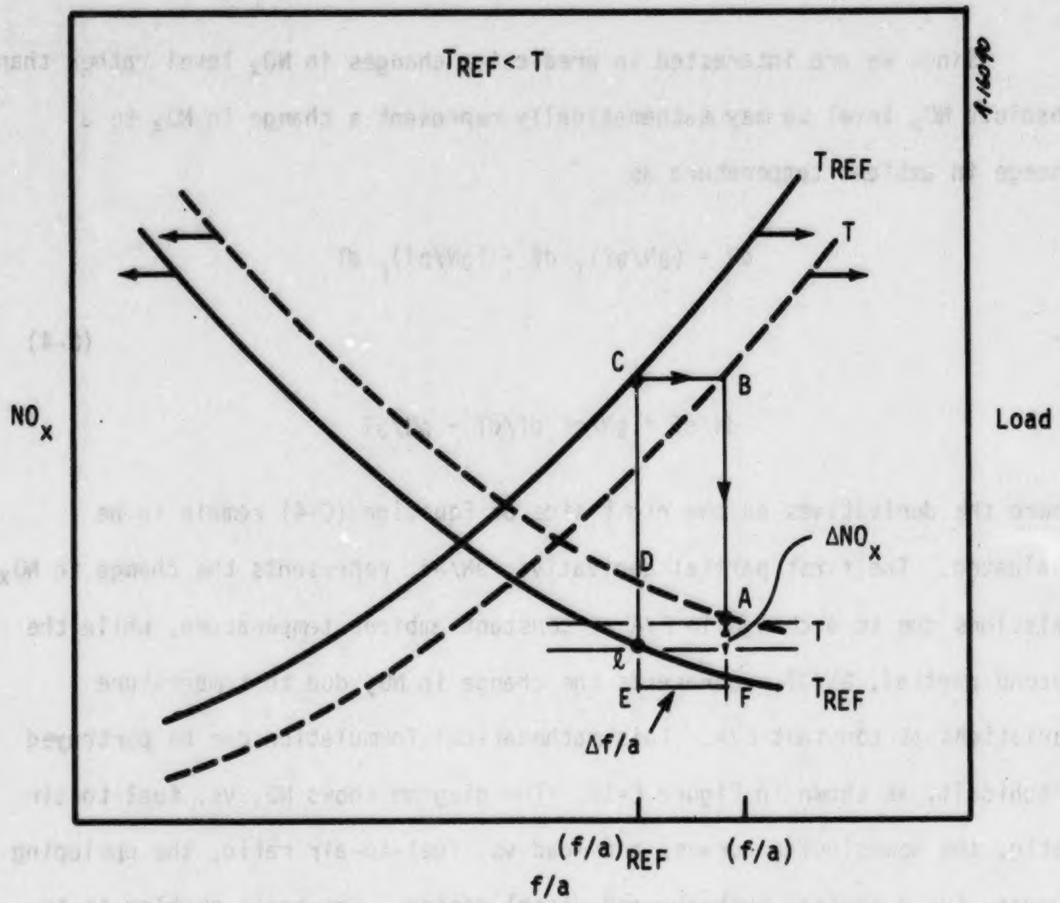

Figure C-17. Relationship of $NO_X$ level and load to f/a ratio for a turbocharged diesel engine operating at two ambient temperatures.

knowing the reference fuel-to-air ratio, one can move from Point D to Point E on the reference temperature curve. Thus, the change in $NO_X$ production resulting in this change of ambient temperature is indicated as $\Delta NO_X$.

The change in $NO_X$ level with F/A for a constant ambient temperature, $\partial N/\partial f$, is presumed known for a given engine or engine type. This relationship can be established from data that can be obtained in the laboratory on a sample engine. In addition, it is assumed that the $NO_X$ vs. F/A plots of Figure C-12 have similar slopes for different ambient temperatures. Therefore, only $NO_X$ vs. F/A data at one ambient temperature are required to evaluate the derivative.

The derivative df/dT, expresses the rate of change of F/A ratio with a change in ambient temperature. Wu and McAulay have derived the following expression relating F/A to ambient temperature (at constant load) for both turbocharged and naturally aspirated engines(72).

$$f = f_R \, (T/T_R)^B$$

and (C-5)

$$B = (1 + n) - (1 - e_i) \, n \, (k - 1/\eta K)$$

where $k = C_p/C_v$, ratio of specific heats

$\eta$ = turbocharger adiabatic compressor efficiency

$n$ = turbocharger exponent from equation $T_A{}^n \, r_c$ = constant where $0.5 < n < 1$ depending on turbocharger compressor pressure ratio

$r_c$ = turbocharger compressor pressure ratio

$e_i = (Tc - Tm)/(Tc - Ta)$

where Tc, Tm are compressor exit and intake manifold inlet temperature, respectively. Then, differentiating Equation (C-5) with respect to inlet temperature we can evaluate the term df/dT in Equation (C-4).

$$df/dT = B(f_R/T_R)\ (T/T_R)^{B-1} = B(f_R/T_R)\ (1 + (\Delta T/T_R))^{B-1}$$

$$\simeq B(f_R/T_R)\ (1 + (B - 1)\ (\Delta T/T_R)) \qquad \text{(C-6)}$$

Since B is always less than 2 and $T_R$ is 70°F = 530°R, a nominal change in ambient temperature of 25°F makes

$$(B - 1)\ T/T_R \leq 1/20$$

Since this term is small compared to unity, we can approximate the derivative by

$$df/dt \simeq B\ (f_R/T_R)$$

Thus, the term from Equation (C-4) that predicts the change in $NO_x$ due to a change in F/A is evaluated as

$$\partial N/\partial f\ df/dt = (\partial N/\partial f)\ (B)\ (f_R/T_R) \qquad \text{(C-7)}$$

An expression is now required to relate the change in $NO_x$ level to a change in ambient inlet air temperature, i.e., the partial derivative $\partial N/\partial T$ of Equation (C-4). This dependence can be evaluated by first relating changes in ambient temperature to changes in flame temperature using the following relationship from Williams' Combustion Theory(73) suggested by Wilson, Muir, and Pellicciotti(74).

$$T_{fl} = T_B + 1/C_p \ \left((Q - L)\ Y_{ox}/i + C_p(T_\infty - T_B)\right)/(1 + Y_{ox}/i) \qquad \text{(C-8)}$$

where $T_{fl} \equiv$ droplet diffusion flame temperature(75)

$T_\infty \equiv$ isentropic compression temperature

$\equiv T_M(CR)^k$

$CR \equiv$ compression ratio of engine

$T_M \equiv$ manifold air temperature, related to $T_A$ by degree of turbocharging and aftercooling

$T_B \equiv$ boiling point temperature of the fuel

$Q \equiv$ heat of combustion per unit mass of fuel

$L \equiv$ latent heat of vaporization of the fuel

$Y_{OX} \equiv$ ambient oxygen mass fraction

$i \equiv$ stoichiometric oxygen-to-fuel ratio (by mass)

$C_p \equiv$ specific heat of the fuel/air mixture

Then, the change in flame temperature is related to a change in $NO_X$ production using the Arrhenius relation

$$dNO_x/dt \propto \exp(-K/RT_{fl}) \ ; \ K/R = 123{,}000^{\circ}R$$

If it is assumed that the rate of $NO_X$ production is independent of time, one can readily integrate the Arrhenius equation to yield the following expression for the ratio of $NO_X$ produced at a given ambient temperature, T, and a corresponding flame temperature, $T_{fl}$, to that at the reference condition.

$$N/N_R = \exp\ \left(K(T_{fl} - T_{flR})/R(T_{fl} \times T_{flR})\right) \qquad \text{(C-9)}$$

The partial derivative can then be approximated in finite difference form by

$$\partial N/\partial T \simeq N(1 - N_R/N)/(T - T_R) \qquad \text{(C-10)}$$

Now by substituting Equations (C-7), (C-9), and (C-10) into Equation (C-4) we obtain:

$$\frac{dN}{dT} = \frac{Bf_R}{T_R}\frac{\partial N}{\partial f} + \frac{N\left\{1 - \exp\left[\frac{K}{R}\frac{(T_{fl} - T_{flR})}{T_{fl} \times T_{flR}}\right]\right\}}{T - T_R} \tag{C-11}$$

A preliminary evaluation of this expression for a turbocharged diesel engine predicts a 1-percent change in $NO_x$ per $1^oF$ change in ambient inlet air temperature. For a constant intercooler effectiveness and turbocharger compressor efficiency, a 1-degree change in ambient temperature corresponds to a 1-degree change in manifold inlet temperature. Therefore, this analytical expression can be checked using $NO_x$ emissions data vs. manifold air temperature (at constant load) for a turbocharged, diesel engine.

Emissions data for the manifold air cooling control (see Figure 4-34 of the draft SSEIS) indicates that this technique produces a 0.1- to 0.3-percent change in $NO_x$ per degree Fahrenheit for diesel engines. Emissions data reported by Ingersoll-Rand (and from Figure 4-34 of the draft), show about a 1-percent change in $NO_x$ per $1^oF$ change in manifold air temperature for turbocharged SI engines(76). Since $NO_x$ emissions from SI engines are more responsive to changes in manifold air temperature, it appears that this analytical approach overestimates the effect of inlet air temperature on $NO_x$ emissions from this turbocharged CI engine. The assumption of a constant $NO_x$ production rate with peak flame temperature may be more valid for SI engines than CI engines owing to differences in the combustion processes.

The approach outlined here has the potential to incorporate many different parameters into a single correction factor. Additional terms could be added for any other operating parameters that have a significant effect on $NO_x$ emissions, either directly or by their effect on F/A or $T_{fl}$. Once the appropriate relations are established, it may be possible to predict changes in $NO_x$ levels as a result of changes in any one of these variables caused by ambient variations.

These goals are somewhat ambitious; therefore, the most logical first step is to gather available data taken at different ambient conditions and apply the method to prove its validity. At this time, however, there is insufficient information for either SI or CI engines to substantiate this approach.

## C.3 REVIEW OF $NO_x$ MEASUREMENT METHODS

Previous studies have indicated that sampling instrumentation and procedures significantly affect emission levels. Furthermore, no one standard procedure has been adopted by all of the manufacturers of stationary IC engines reporting emissions. That is, eight large-bore engine manufacturers who reported emission data used either chemiluminescent (CL) or nondispersive infrared (NDIR) and ultraviolet (NDUV) instruments and one of four emission measurement procedures (SAE, EPA, EMD, or DEMA).[2/] Therefore, this section will present a detailed discussion of the instrument and sample acquisition practices used by the engine manufacturers. The purpose of this discussion is to identify possible variations in exhaust emission levels attributable to particular measurement equipment and/or procedures. Specifically, this section will:

- Establish which instruments, sampling trains, and procedures were used by each manufacturer who reported $NO_x$ emissions
- Summarize the potential sources of uncertainty relating to each measurement practice
- Evaluate the variability among manufactuers' emissions data due to

the following discussion will begin by illustrating typical measurement uncertainties in present sampling practices. This analysis will show that significant uncertainties in measurements can arise due to differences in both instrument and sampling procedures. Therefore, both of these sources of error will be discussed as they apply to the instruments and procedures used by the

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[2/] Alco, Cott, Cooper, DeLaval, Electro Motive, Ingersoll-Rand, Waukesha, and White Superior.

eight large-bore engine manufacturers. Then these practices will be compared and the uncertainty in each manufacturer's $NO_x$ measurements estimated.

### C.3.1 Previous Studies of Exhaust Measurement Variation

Before evaluating each manufacturer's sampling practice, it is of interest to examine the present "state-of-the-art" of internal combustion engine exhaust measurements of $NO_x$. Three recent studies have been conducted to compare the measurements of $NO_x$ made by different laboratories from the same emission source. In two of these studies, $NO_x$ measurements were made, simultaneously, in the same laboratory using identical procedures. In the other study, the same emission source was sent to each laboratory. These studies illustrate the magnitude of emission variations attributable to instruments and sampling procedures. They also indicate the reproducibility of emissions measurements, both within a laboratory and among different laboratories. The results of these studies will serve as a basis for comparing potential data variations due to the measurement practices of the eight large-bore engine manufacturers considered in the current study.

A series of cooperative emissions tests was conducted by the Coordinating Research Council (CRC) to evaluate measurement methods used to analyze diesel exhaust emisions from truck-size engines. In Phase III of this program, six laboratories sent sampling teams to one location to make simultaneous measurements (using NDIR analyzers) of a multicylinder engine(77). The engine used in this study was a six-cylinder, 300-cubic-inch, four-stroke, direct-injection diesel. The procedures that were used during this test program to measure NO, CO, and $CO_2$ evolved into SAE Recommended Practice J-177(78). In Phase IV of that program, the same engine was circulated to 15 laboratories to: (1) verify that the generally good agreement of Phase III emission levels were the result of improved sampling procedures, and (2) obtain NO/$NO_x$ data using CL analyzers(79).

Table C-84(80,81) shows the results of these two cooperative tests. Although the Phase III variations appear reasonable, the Phase IV results indicate poor agreement among the laboratories. These larger uncertainties were attributed to poor calibration procedures (span gases out of spec, instruments not calibrated) and possibly some variation in engine performance. In addition, it was noted that the average NO concentration measured at rated load by the CL analyzers was approximately 23 percent lower than that measured by NDIR.

A more recent cooperative test program conducted by the CRC evaluated EPA's revised heavy-duty diesel engine NO/$NO_x$ measurement methods and instrumentation(82). Six participants made a series of NO/$NO_x$ measurements on a multicylinder engine, simultaneously, and produced the range of uncertainty shown below. A range is shown because the results were analyzed in three different groupings: (1) all data, (2) those which remained after eliminating questionable results from participants who encounter sampling problems, and finally, (3) those which were left after excluding both questionable data and results obtained from instruments with long sample transfer times.

|  | 13-mode (g/hp-hr) | Standard Deviation (g/hp-hr) | Standard Deviation (Mean, %) |
|---|---|---|---|
| NO | 8.03 - 8.21 | 0.42 - 0.29 | 5.9 - 3.5 |
| $NO_x$ | 8.05 - 8.16 | 0.30 - 0.17 | 3.8 - 2.1 |

All but one of the participants used a CL analyzer, and the results from the one NDIR analyzer (a new reduced interference design with a $NO_x$ converter)

## TABLE C-84. ACCURACY OF $NO_x$ MEASUREMENTS(80,81)

| Source | Phase III<br>6 Labs, Simultaneous Measurements | | Phase IV<br>15 Labs, Round Robin | |
|---|---|---|---|---|
| | Standard Deviation as % of Average | Spread in Data as % of Average | Standard Deviation as % of Average | Spread in Data as % of Average |
| NO: | | | | |
| NDIR | 5 | 10 | 8 | 27 |
| CL | — | — | 36 | 15 |
| $NO_2$: | | | | |
| NDUV | — | — | 21 | 39 |
| CL | — | — | 88 | 166 |

were equivalent to those from the CL analyzers. Thus, based on simultaneous tests using the same instruments, sampling practices, and emissions sources, uncertainties in emission measurements among laboratories ranged from 3 to 6 percent for NO, and 2 to 4 percent for $NO_x$. In addition, the repeatability within a laboratory ranged from 2 to 7 percent for NO and 1 to 3 percent for $NO_x$.

In contrast to the above program, the eight large-bore engine manufacturers used one of four test procedures (SAE, EPA, DEMA, or EMD) and either CL or NDIR/NDUV instruments. Furthermore, the NDIR's were not reduced interference designs. Hence, their data are likely to vary more due to measurement practices than the truck engine results. The following discussion will first examine the sources of such variations from differences in both instruments and sampling practices. Then an attempt will be made to suggest uncertainty bounds for the emisions data from each manufacturer.

### C.3.2 Variations in $NO_x$ Emissions Related to the $NO_x$ Instrument

Since the development of commercial chemiluminescent analyzers, (1971), various studies have been conducted to compare their operation with the already established NDIR analyzer. All of these studies have shown the NDIR analyzer to record consistently higher levels of NO than the CL analyzer for a given source(83-85). Three of the large-bore engine manufacturers reported $NO_x$ emissions using NDIR's and the other five used CL analyzers.

Table C-85 shows the $NO_x$ instruments used by each of the manufacturers. Scott Research Laboratories made the emission measurements for Ingersoll-Rand and White-Alco, since neither manufacturer presently owns emission measurement equipment. Note that an electrochemical instrument was used by Shell Oil Research in 1971 to measure $NO_x$ from one Cooper-Bessemer

TABLE C-85. INSTRUMENTS USED BY MANUFACTURERS TO MEASURE $NO_x$ EMISSIONS

| Manufacturer | CL (with Thermal Converter) $NO/NO_x$ | NDIR/NDUV $NO, NO_2/NO_2$ | Electrochemical $NO/NO_2$ |
|---|---|---|---|
| Colt | Thermo Electron 10A | | |
| Cooper Energy | Thermo Electron 10A | NO: Beckman IR 18A | Envirometrics NS-200 |
| DeLaval | Scott 125 | | |
| Ingersoll-Rand[a] | Scott 125 | | |
| Waukesha | | NO: Horiba AIA-2 | |
| White Superior (Div. Cooper Energy) | Scott 125 | | NO: Dynasciences NX-130<br>$NO_2$: Envirometrics NS-200 |
| White-Alco[a] | Scott 125 | | |
| GMC/EMD | | NO/Beckman 315B<br>$NO_2$/Beckman 255 | |

[a] Measurements made by Scott Environmental Technology Laboratory.

and one Ingersoll-Rand engine. None of the manufacturers has used this instrument since then; therefore, no attempt will be made to correlate those emissions with instrument differences. Only Waukesha and GMC/EMD continue to measure $NO_x$ emissions with NDIR analyzers, and Waukesha has recently acquired a CL analyzer. Thus, all but one manufacturer will be using CL instruments in the future. In the following paragraphs, the principle of operation of the NDIR and CL instruments and associated sources of error will be briefly reviewed. Then the results of NDIR/CL comparisons will be summarized to seek a method of expressing emissions on an "equivalent instrument" basis.

### C.3.2.1 NDIR Instrumentation

The NDIR instrument was introduced nearly 20 years ago and has continued to be used widely as a CO and $CO_2$ detector. In addition, it was used extensively to detect NO, HC, and $SO_2$, but other methods are not supplanting it for these species. Its principle of operation depends upon absorption of infrared radiation by the gaseous sample. Figure C-18 illustrates a typical NDIR instrument. Built-in optical and gaseous filters are used to produce a narrow infrared beam band width to compensate for interference (absorption) by other constituents.

Despite these precautions, water vapor and $CO_2$, to some extent, may cause positive interferences (high readings) even though refrigerant and chemical driers are used to remove water vapor. Desiccants, however, have been found to cause significant interferences as well as water vapor (see later discussion and Section C.3.3). In addition to these problems, the response of the NDIR instrument to the specie of interest is nonlinear in some instrument designs, necessitating a carefully constructed calibration curve using at least four, and preferably six to eight, calibration gases.

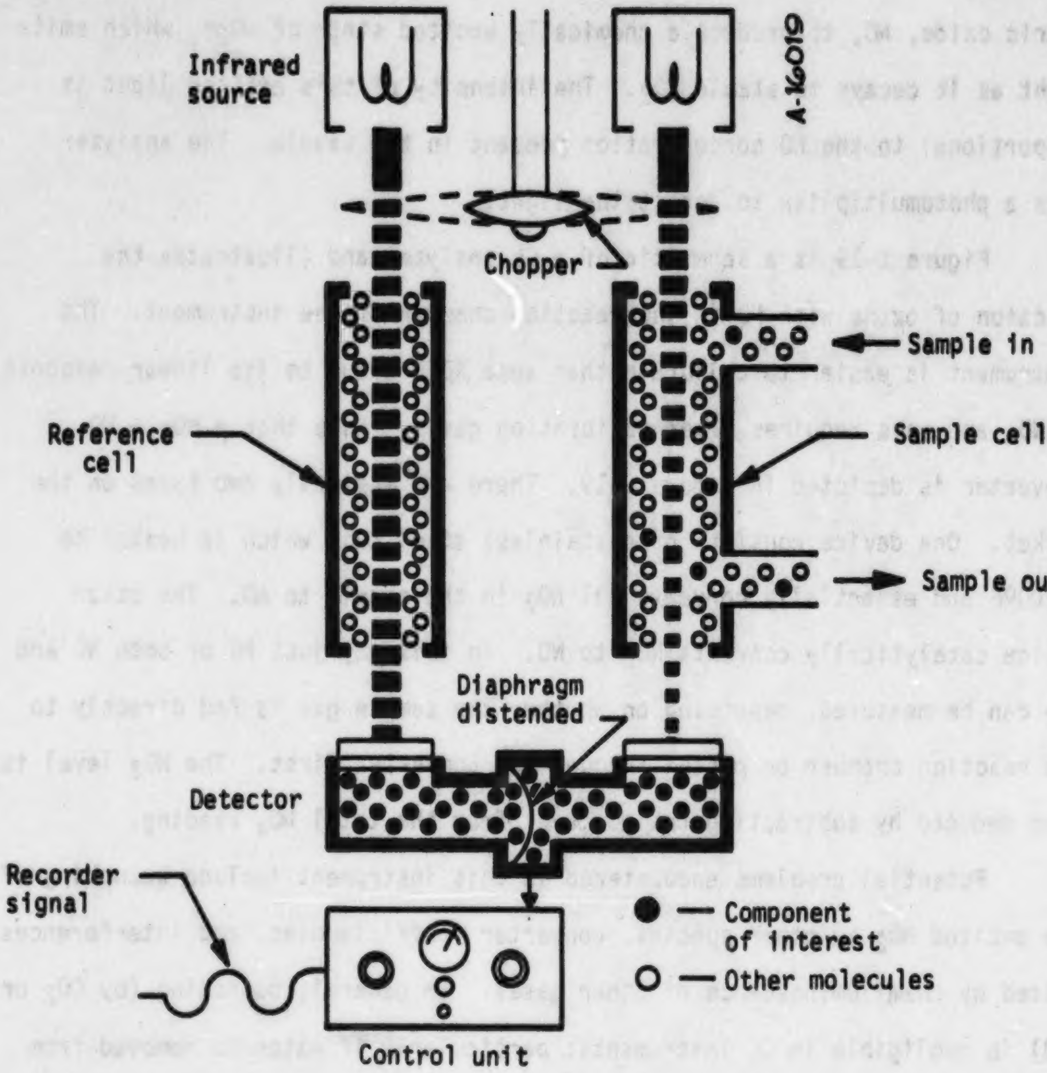


Figure C-18. Nondispersive infrared absorption analyzer.

### C.3.2.2 Chemiluminescent Analyzers

Chemiluminescent analyzers, in contrast to NDIR instruments, have been developed only recently (1971) for source sampling. Nevertheless, the CL instrument has gained increasing application for the measurement of $NO_x$ ($NO + NO_2$) and NO. In this type of instrument ozone, $O_3$, is reacted with nitric oxide, NO, to produce a chemically excited state of $NO_2^*$, which emits light as it decays to stable $NO_2$. The intensity of this emitted light is proportional to the NO concentration present in the sample. The analyzer uses a photomultiplier to detect the light.

Figure C-19 is a schematic of a CL analyzer and illustrates the reaction of ozone with NO in the reaction chamber of the instrument. The instrument is easier to calibrate than some NDIR's due to its linear response to NO, and thus requires fewer calibration gases. Note that a $NO_2 \rightarrow NO$ converter is depicted in Figure C-19. There are basically two types on the market. One device consists of a stainless steel tube which is heated to $\approx 1200°F$ and essentially converts all $NO_2$ in the sample to NO. The other device catalytically converts $NO_2$ to NO. In this way just NO or both NO and $NO_2$ can be measured, depending on whether the sample gas is fed directly to the reaction chamber or passed through the converter first. The $NO_2$ level is then deduced by subtracting the NO level from the total $NO_x$ reading.

Potential problems encountered in this instrument include quenching of the excited $NO_2$ by other species, converter inefficiencies, and interferences caused by chemiluminescence of other gases. In general, quenching (by $CO_2$ or $H_2O$) is negligible in CL instruments; particularly if water is removed from the sample before analysis and low pressure (vacuum) reaction chambers are used to reduce quenching by $CO_2$. Nevertheless, some quenching problems have been observed during measurement of fuel-rich automotive exhausts. Quenching

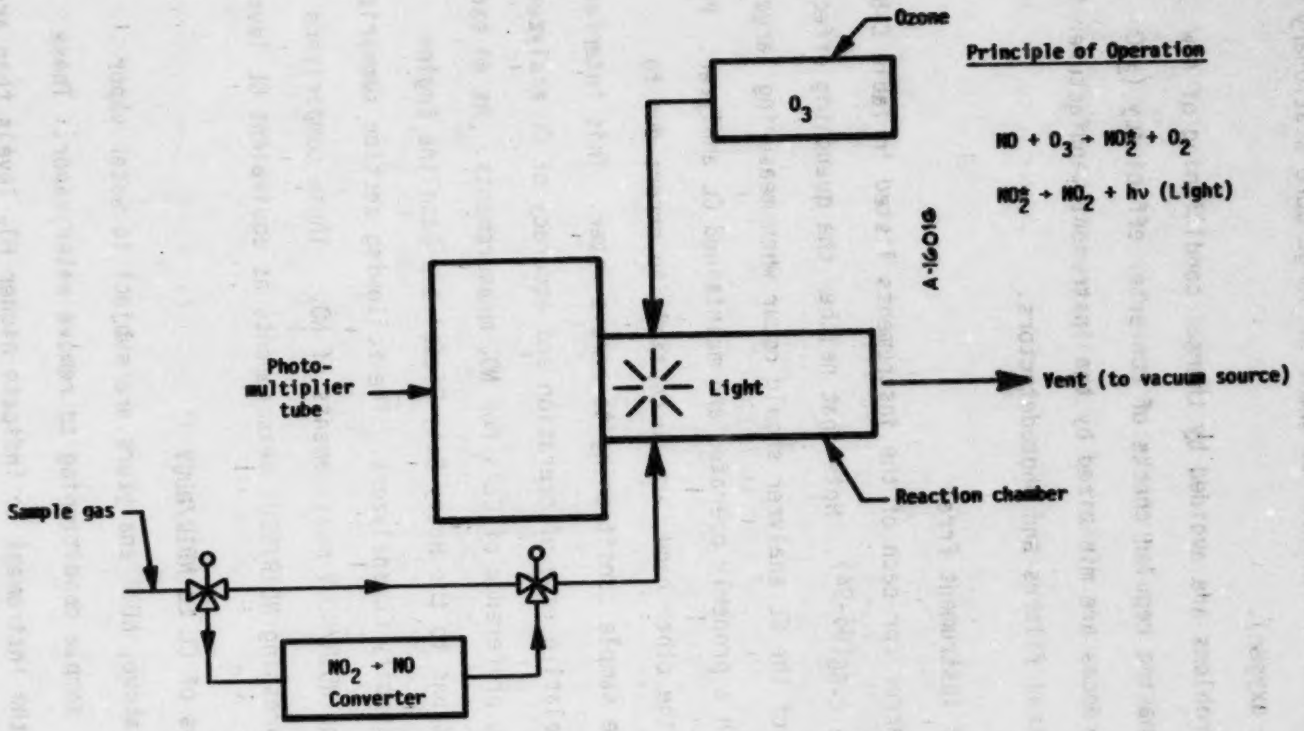


Figure C-19. Chemiluminescent analyzer.

effects, however, have not been observed during measurements on truck-size diesels, whose exhausts are similar to those of large bore stationary engines (i.e., ≈15 percent oxygen).

Converter problems are avoided by thermal conditioning of new converters and by making regular checks of converter efficiency ($\geq$90 percent). Interferences are minimized by the instrument manufacturer through the choice of spectral filters and photodetectors.

### C.3.2.3 Sources of Instrument Error

Sources of error for each of the instruments listed in Table C-85 are summarized in Table C-86(86-94). Note that neither the quenching effect nor the converter problems of the CL analyzer should occur when measuring large-bore engine exhausts with a properly operated and maintained CL analyzer. Present NDIR analyzers, on the other hand, are susceptible to errors due to interference despite sample conditioning to remove water. This interference, combined with the relative ease of operation and accuracy of CL analyzers, has led to an increasing preference of CL's for $NO_x$ measurements. As an example, EPA's proposed revisions to the Heavy Duty Diesel and Gasoline Engine Sampling Procedure specify CL analyzers. The following section summarizes comparisons of CL and NDIR/NDUV measurements of $NO_x$. These comparisons will be the basis for expressing NDIR/NDUV measurements as equivalent CL levels.

### C.3.2.4 Correlations of CL to NDIR/NDUV

As discussed above, NDIR analyzers are subject to water vapor interference despite sample conditioning to remove water vapor. These interferences cause the instrument to indicate higher $NO_x$ levels than are actually present. A comprehensive study conducted by TRW (with Scott

TABLE C-86. SOURCES OF $NO_x$ INSTRUMENT ERROR

| Chemiluminescent, $NO/NO_2$ | NDIR, NO (86,87) | NDUV, $NO_2$ (88) | Electrochemical $NO/NO_2$ (89) |
|---|---|---|---|
| • Quenching of CL by third body reactions[a] ($CO_2$, $H_2O$)<br>• $NO_2$/NO converter[b]<br>— $NO_x \rightarrow N_2$ at high temperature, low $O_2$, high CO and HC<br>— $NO_2 \rightarrow NO$ incomplete due to improper converter conditioning<br>— $NH_3 \rightarrow NO, NO_2$ at high temperature especially with some catalytic converters in rich exhausts<br>— $NO_2 \rightarrow NO$ at low temperatures, low O2 | • Water vapor interference causes higher $NO_x$ readings. Also $CO_2$ interference to some extent; optical filters and water removal minimize interferences<br>• Use of chemical driers lead to $NO_2$ adsorbtion and $NO_2 \rightarrow NO$ conversion<br>• Calibration curve non-linear, requires frequent checks using 4 to 6 calibration gases | • Interference caused by carbon particles and other species, results in a drifting, high reading. | • Not much application in source sampling<br>• $SO_2$ interferences<br>• Nonlinearity in 30 to 300 ppm range<br>• Susceptible to error due to temperature changes |

[a] Not present in properly designed instruments (90-92).

[b] Condition CL converter to avoid error; perform regular converter efficiency. Most of these errors occur in rich exhausts unlike the lean exhausts (~15 percent oxygen) of large bore engines (93-94).

Research Labs) using constant volume sampling (CVS) of automotive exhausts determined that

$$(NO)_{NDIR} = 1.07\ (NO)_{CL} - 4.2 \qquad \text{(C-12)}$$

where (NO) denotes NO concentration in ppm.(95) Both analyzers had similar water removal systems (condenser and chemical driers); hence, the difference was due to a gas interference only. In addition to this finding, the study established that chemical driers promoted $NO_2 \rightarrow NO$ conversion. Comparison of an NDIR with water removal to a CL without water removal indicated that

$$(NO)_{NDIR} = 1.12\ (NO)_{CL} + 8.4 \qquad \text{(C-13)}$$

Furthermore, a comparison of two CL analyzers, one ($CL_B$) preceded by a water trap and chemical drier and the other ($CL_A$) without either, indicated that

$$(NO_x)_{CL_B} = 0.873\ (NO_x)_{CL_A} + 6.6 \qquad \text{(C-14)}$$

Thus, the net effect of the drier and water trap is to destroy about 12 percent of the $NO_x$ in the sample.

The results of this study and others that corroborate these effects are summarized in Table C-87(96-100). The bias of NDIR analyzer ranges from 3 to 30 percent higher based on these studies. As this table indicates, the particular correction that applies to any one of the three manufacturers that used an NDIR analyzer (and NDUV for $NO_2$) will depend on their sampling setup and procedures (e.g., type and location of water trap, and the $NO_2$ present). Therefore, the appropriate correction for NDIR instrument bias will be identified after a discussion of each manufacturer's sampling practices.

## TABLE C-87. COMPARISON OF CL AND NDIR, NDIR/NDUV MEASUREMENTS

| Source | Comparison | Comments |
|---|---|---|
| TRW Study (96)<br>(Reference 5) | $NO_{xCL}/NO_{xNDIR/NDUV}$ = 0.82 | No water removal from CL sample. |
| | $NO_{xCL}/NO_{xNDIR/NDUV}$ = 0.75 | Water trap and drier used for both CL and NDIR. |
| Clemmens Memo<br>(97) | $NO_{xCL}/NO_{xNDIR}$ = 0.9<br>($NO_{CL}/NO_{NDIR}$ = 0.88) | Used TRW comparison for NO (no water removal for CL) and assumed that $NO_2$ was 2 percent (on the average) of total $NO_x$ in a diesel exhaust. |
| SAE 750204<br>Phase IV<br>(Reference 98) | $NO_{CL}/NO_{NDIR}$ = 0.77 | Result questionable due to other sampling errors. |
| SAE 730213<br>(Reference 99) | $NO_{xCL}/NO_{xNDIR/DUV}$ = 0.94 | None of eight large-bore manufacturers used DUV[a] instrument. |
| CRC Report #487<br>(Reference 100) | $NO_{xCL}/NO_{xNDIR}$ = 0.985 | NDIR instrument is "third" generation, low interference analyzer, not comparable to NDIR's used by large-bore manufacturers. |

[a] Dispersive Ultraviolet

### C.3.3 Variations in $NO_x$ Levels Related to Sampling Procedures

In addition to the choice of instrumentation, differences in sampling procedures can cause variations in reported emission levels. This was clearly illustrated in Table C-84 for the Phase IV cooperative tests where 15 laboratories made measurements of an identical source. Since four basic sampling procedures (DEMA, SAE, EMD, EPA) were used by the eight large-bore engine manufactuers, some variation between their reported emission levels can probably be attributed to differences in their sampling practices. The sources of these uncertainties will be identified and the four sampling procedures compared and evaluated in this section.

#### C.3.3.1 Sources of Sampling Error

The two major problems encountered in measuring engine exhausts are: (1) chemical changes that occur during transfer of the sample to the analyzer, and (2) error due to improper operation of the analyzer. Figure C-20 is a simple schematic of an engine sampling system. In transferring the exhaust gas sample to the analyzer, care must be taken to ensure that all of the $NO_x$ ($NO + NO_2$) or NO (when only an NDIR is used) originating from the engine exhaust reaches the analyzer. For these reasons sample lines are heated to prevent condensation or kinetic conversion of constituents. High sample flowrates (short sample residence times) are maintained to minimize sample degradation during its transfer from the engine to the analyzer, and water removal devices are employed to minimize instrument interferences from water vapor. Similarly, it is important that the analyzer be calibrated and all components, such as $NO_2 \rightarrow NO$ converters (on CL instruments), be functioning properly. Therefore, adequate analyzer specifications and calibration procedures are essential for accurate emission measurements.

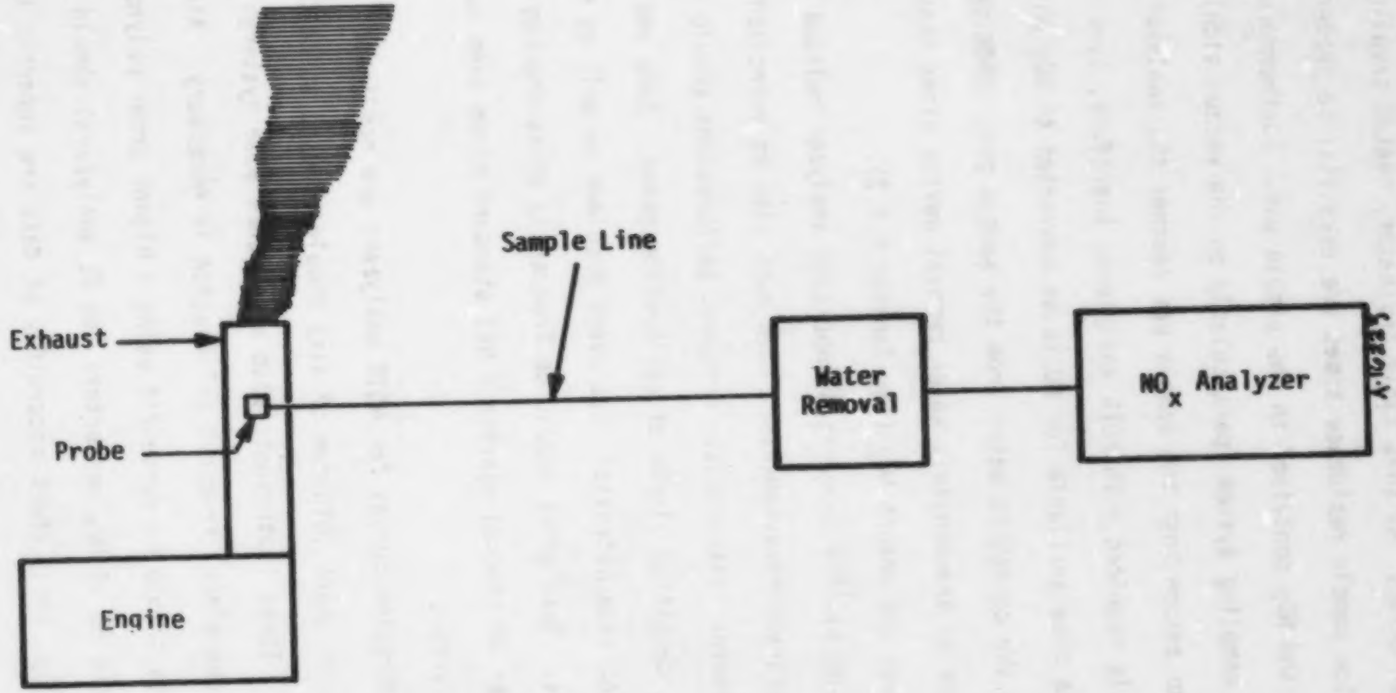


Figure C-20. Schematic of exhaust sampling system.

Table C-88(a)(101-107) presents a more detailed summary of sources of sample transfer error. As this table indicates, heated sampling lines coupled with low sample residence times are essential to preserve the initial amounts of NO and $NO_2$ contained in the sample gas. Furthermore, leak checks of the entire sampling system (particularly on the vacuum side) should always be performed to assure that the sample gas reaches the analyzer undiluted. Water removal is required with NDIR analyzers; therefore, care must be taken to minimize the time available for NO to be converted to $NO_2$ which can then be absorbed by the condensed water from the sample gas. Chemical driers (desiccants) are an unacceptable water removal device since they promote the $NO_2 \rightarrow NO$ reaction and absorb $NO_2$ (see Section C.3.2).

Table C-88(b)(108) summarizes important analyzer related procedures required for accurate measurements. Care must also be exercised in operating emission measurement instruments. Frequent calibrations should be performed using accurate, certified blends of calibration gases. Zero and span checks also serve to indicate potential instrument problems as well as necessary gain adjustments. Span gases should be frequently cross-checked with calibration gases or checked against a NBS standard since even certified gases can be in error.

Some calibration curves for NDIR analyzers are nonlinear; therefore, several calibration gases (minimum of six) should be used to calibrate the instrument(109). These instruments also experience some hysterysis after sitting unused; therefore, frequent calibration is necessary. Also, calibration points should be curve-fit using a higher order polynomial. In addition, efficiencies of $NO_x$ converters (on CL analyzers) should be checked regularly. Finally, strip chart recordings of data are superior to visual

## TABLE C-88(a). SOURCES OF SAMPLING ERROR: THE SAMPLE TRANSFER

| Sampling System | | |
|---|---|---|
| **Source** | **Error** | **Correction** |
| ● Unheated sampling line | NO → $NO_2$; $NO_2$ adsorbed during water removal: Observed 40-percent loss of NO when sample residence times were 35-40 sec (101) | Heat line to 375°F |
| ● Long sample residence times | Heated Line: $NO_2$ adsorbed during water removal: 6-7% $NO_x$ loss for system response times[a] greater than 25 sec (102)<br>Cold Line: 12-percent $NO_x$ loss for sample residence time of 13 sec (103) | Use short sample line and/or flowrates to limit system response to 15 sec or less. |
| ● Sample line connections and fittings | Leaks that dilute sample gas: Observed 25-percent loss of $NO_x$ due to prefilter leak (104) | Leak check system before testing |
| ● Water removal | | |
| — Refrigerant | $NO_2$ adsorbed in condenser; (105) | CL: Locate condenser after converter.<br>NDIR: Maintain high sample flow through condenser, remove $H_2O$ as it forms. |
| — Chemical drier (desiccant) | $NO_2$ → NO and drier "eats" $NO_2$. Negative or positive errors with cold lines (SAE procedure) depending on how drier conditioned (106, 107) | Do not use chemical driers |

[a] System response time = sample residence time + instrument response time

TABLE C-88(b). SOURCES OF SAMPLING ERROR: INSTRUMENT RELATED

| Instrumentation | | |
|---|---|---|
| **Source** | **Error** | **Correction** |
| ● Instrument drift, ozone shortage (CL), plugged capillary | Low $NO_x$ readings | Calibrate instrument frequently. Zero and span before and after each measurement for all instrument ranges. Instrument should meet minimum performance criteria. |
| ● Converter malfunction (CL) | Not all $NO_2$ converted to NO, results in low $NO_x$ reading also possible for NO to exceed $NO_x$ levels | Perform converter checks regularly. Use known standard to check converter efficiency. |
| ● Visual rather than stripchart readings | Analyzer meter only accurate to ±3 percent. Visual averaging less consistent than chart averaging (108) | Use stripchart, average levels over an interval for which steady state conditions exist. |
| ● Calibration and span gases | Change of constituents with time, or erroneous certification | Use certified gases of specified blends. Cross-check span and calibration gases. |

analyzer meter readings since a permanent record is produced, and emission levels can be averaged over a time interval more accurately and consistently.

### C.3.3.2 Sampling Procedures: EPA, DEMA, SAE, EMD

Having completed this brief review of sources of sampling error, we can examine each of the four measurement practices presently used by the eight large-bore engine manufacturers. Table C-89 is a comparison of the EPA, DEMA, SAE, and EMD emission measurement practices(110-113). The information for the EPA procedures is based on the revised EPA Heavy Duty Diesel Emissions Regulations as proposed in the Federal Register, Volume 41, No. 101, May 24, 1976. This revised procedure requires CL analyzers for NO/$NO_x$ measurement, establishes instrument and calibration specifications, and defines sample transfer configurations and practices.

This comprehensive EPA regulation was recently verified through cooperative testing by both manufacturers of mobile diesel engines and the EPA-Ann Arbor Mobile Sources Laboratory(114). The six participants in this program (see Section C.3.1) made essentially equivalent $NO_x$ measurements for a series of eight tests. Two participants, however, did experience a small, but consistent, NO/$NO_x$ crossover (NO levels greater than $NO_x$ levels). Nevertheless, the standard deviation of measured $NO_x$ levels was generally small, ranging from 2 to 4 percent (of the mean level). Therefore, the EPA practice will be the basis of comparison for the other measurement practices.

The potential sources of measurement uncertainty of the other three procedures (DEMA, SAE, and EMD) are summarized in Table C-90. The sampling trains for these three procedures, as well as the EPA setup, are illustrated in Figure C-21. The DEMA procedure specifies a CL analyzer and the SAE/EMD

## TABLE C-89. COMPARISON OF SAMPLING PRACTICES

- Test Procedure

DEMA: No time between modes specified
EPA: 5 minutes
SAE: 20 minutes
EMD: 30 minutes

DEMA: Data recorded and averaged over 10 minutes
EPA: Data recorded over last 2 minutes
SAE: Data recorded over last 5 minutes
EMD: Data recorded over last 3 minutes

- Instrument Calibration — DEMA and EPA are CL; SAE is NDIR; EMD is NDIR/NDUV

DEMA: Analyzers calibrated semimonthly. No details of calibration given.
EPA: Check $NO_x$ converter once per week (must be at least 90-percent efficient); leak check system, calibrate analyzer, check sample line residence time, and quench check every 30 days
SAE: Calibrate monthly
EMD: Checked and calibrated monthly, semiannually and annually

DEMA: Number of calibration gases not specified
EPA: Span analyzer with calibration gases having nominal concentrations of 30, 60, and 90 percent of full scale concentrations
SAE: Use calibration gases that are 25, 50, 75, and 100 percent of instrument range used.
EMD: Use 4 calibration gases

DEMA: No specifications for calibration or span gas blends and dilutents
EPA: Blends and dilutents specified and accurate within 2 percent of true concentration or traceable within 1 percent of NBS blends. Span gas traceable to within 1 percent of calibration gas.
SAE: ±2 percent accuracy on gas analysis certification
EMD: ±2 percent accuracy blends and dilutents defined by EMD

DEMA: No analyzer specifications required
EPA: Specifications for response-time, precision, noise, zero and span drift, and linearity
SAE: Accuracy should be ±2 percent full-scale deflection or better
EMD: Same as SAE

TABLE C-89. Concluded

- Sample Transfer

DEMA: Stainless steel (S.S.) probe, configuration not specified
EPA: Specifies number and size of holes in S.S. probe
SAE: S.S. probe, no configuration specified
EMD: S.S. probe, configuration specified

DEMA: Sampling line material not specified
EPA: Stainless steel, teflon, or proven inert
SAE: Stainless steel or teflon after exhaust cooled
EMD: Stainless steel

DEMA: Line heated to 200°F for $NO_x$ (375°F for $HC_T$)
EPA: 375°F + 18°F - 9°F
SAE: Unheated
EMD: Unheated

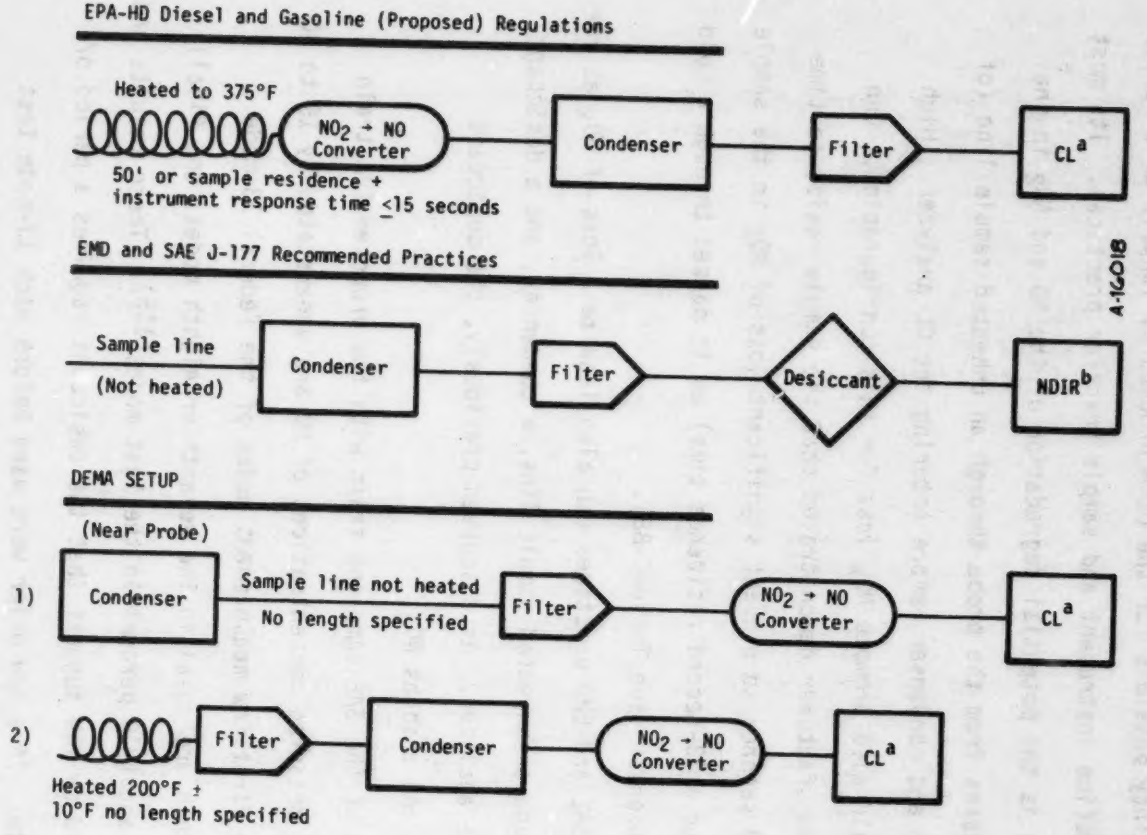
DEMA: No sample line length specified
EPA: <50 feet or sample line residence and instrument response time is less than or equal to 15 seconds
SAE: No length specified
EMD: No length specified

DEMA: Chemical driers can be used
EPA: Not allowed
SAE: Refrigerant and chemical drier specified
EMD: Same as SAE

DEMA: Recommend water removal at probe or at analyzer before $NO_2 \rightarrow NO$ converter
EPA: Condenser at analyzer after converter
SAE: Condenser and drier before analyzer
EMD: Same as SAE for NDIR. NDUV does not use water removal.

## TABLE C-90. SOURCES OF UNCERTAINTY FOR DEMA, SAE, EMD EMISSION PRACTICES

| DEMA (CL) | EMD (NDIR/NDUV)/SAE (NDIR) |
|---|---|
| • Unheated sampling lines permitted<br>• No specification of sample residence time in sampling line, or system response time<br>• Leak checks are not specified<br>• Water removal device can be located at analyzer, but before $NO_2 \rightarrow NO$ absorption in water trap<br>• Chemical driers permitted<br>• No instrument specifications<br>• Converter efficiency checks not specified<br>• No calibration procedures specifed<br>• Calibration and span gas specifications not defined | • Unheated sample lines<br>• No sample residence or system response time specified<br>• Leak checks not specified<br>• Allow chemical drier<br>• Calibration procedures not specific (e.g., what constitutes out-of-calibration, how calibration points are curve fit, etc.)<br>• Calibration and span gas blends and dilutents not specified by SAE. EMD has "own" specifications. |



[a] CL – Chemiluminescent analyzer

[b] NDIR – Nondispersive infrared analyzer

Figure C-21. Sampling practices.

procedures require an NDIR instrument. The EMD practice also utilizes an NDUV analyzer, in series after the NDIR, to measure $NO_2$ in the sample.

Table C-90 clearly indicates that there are several sources of measurement uncertainty possible for the DEMA procedure, largely as a result of its failure to define instrument and sample transfer practices. Its most obvious shortcoming is the potential degradation of the NO and $NO_2$ in the sample gas as it passes from the probe through an unheated sample line (of unspecified length) and condenser before entering the CL analyzer. High residence times would also promote $NO_x$ loss for this configuration. One large-bore engine manufacturer demonstrated that the sample residence time must be less than 3 seconds to prevent significant loss of $NO_2$ in the sample (12-percent loss for a 13-second residence time) as it passed through a cold sample line and condenser (see Table C-88).

Similarly, SAE and EMD practices can also lead to a loss of $NO_x$ as the sample passes through an unheated sample line, a condenser, and a desiccant before reaching the analyzer. As discussed previously, the desiccant converts $NO_2$ to NO and absorbs $NO_2$.

Comparisons of the SAE sampling train with the proposed EPA train indicated that SAE measured concentrations of NO are understated by 15 to 40 percent during the first few measurement modes of the federal 13-mode composite cycle, then are equal in the seventh or eighth modes, and finally are overstated by about 10 percent in the last modes(115). These results and the study conducted by TRW suggest that the desiccant requires a period of time to equilibrate. If a new drier were used before each 13-mode test (permissible under present EPA regulations), NO levels would tend to be low (since levels would be understated until the drier had equilibrated, possibly not until the last, rated speed, low power modes were being measured). If the drier was not replaced, or if it was preconditioned, the NO levels would tend to be overstated. This is probably the case for those large-bore engine manufacturers who use driers, since their engines are typically operated about an hour before they are stabilized and measurements are begun.

The EMD procedure is similar to the SAE procedure with the exception that both NO and $NO_2$ are measured using NDIR and NDUV analyzers. Results from the TRW study$^{(116)}$, based on automotive exhausts, indicated that NDIR/NDUV $NO_x$ measurements were about 20 percent higher than CL readings (no water removal for CL sample). Apparently, the positive biases (interference and drier) more than offset potential $NO_x$ loss due to the cold sampling in that study. In addition to the positive NDIR biases, NDUV analyzers have a history of stability problems, usually manifested as a drifting (high) reading. One study attributed this error to interference due to carbon particle buildup$^{(117)}$.

### C.3.3.3 Summary and Conclusion of DEMA, SAE, and EMD Practices

In summary, it appears that the DEMA practices will generally lead to negative errors due to $NO_x$ loss in transporting the sample from the engine to the analyzer. At this time the absolute uncertainty associated with this practice is unknown relative to the EPA procedure. Limited results (see Table C-87) suggest this error could be as much as -10 to -15 percent depending on the amount of $NO_2$ in the sample (estimated to be 2 to 10 percent of the total $NO_x$) and the extent that NO is converted to $NO_2$ during the sample transfer.

The positive bias of the SAE procedure gives a 12-percent error in the NO reading. The EMD procedure, which measures both NO and $NO_2$ causes a 20-percent error in the $NO_x$ reading. Nevertheless, both procedures could experience as much as a -40-percent error in the NO or $NO_x$ reading if the chemical drier has not stabilized. Thus, the uncertainties in $NO_x$ measurements for the SAE and EMD practices appear to be significantly greater than the DEMA practice.

Finally, the EPA procedure defined for the heavy-duty diesel and gasoline engines appears to have minimized the various sources of measurement error and should give more accurate and consistent $NO_x$ readings. Therefore, each of the manufacturer's sampling practices will be evaluated in the following section by comparing it to the EPA procedure.

### C.3.3.4 Comparison of Manufacturer's Measurement Uncertainty

Table C-91 summarizes each manufacturer's practice in terms of procedures that could lead to measurement error. Note that three of the four DEMA manufacturers using cold sample lines have located their water traps near the analyzer instead of at the engine as is recommended in the DEMA practice. Therefore, this setup will promote additional loss of $NO_x$ in the exhaust sample. In addition, Colt and Waukesha have relatively high sample residence times which will also promote $NO_x$ loss.

The potential errors in measurement resulting from these practices are estimated in Table C-92(118-122) relative to the EPA procedure. (Blanks appear in Table C-92 for items which do not apply to a particular manufacturer.) These uncertainties are depicted for each manufacturer in Figure C-22. Errors due to system leaks are not included in the figure since this error cannot be generalized. Also, the uncertainty bands were constructed assuming that errors of the same sign were additive. Note that

## TABLE C-91. LARGE-BORE ENGINE MANUFACTURER'S SAMPLING PRACTICE

T-88

| Practice Manufacturer | Sample Line |  |  | Water Removal |  |  | Data Recording | Leak Check | Converter Eff. Check (CL) |
|---|---|---|---|---|---|---|---|---|---|
|  | Heated | Length, ft | Residence Time, sec | Refrig | Drier | Location |  |  |  |
| DEMA (CL) |  |  |  |  |  |  |  |  |  |
| Colt | No | 20 | 14 | Yes |  | Before Analyzer | Stripchart | Yes | 2/yr |
| Cooper | No | 20 | 3 | Yes |  | Before Analyzer | Visual | No | No |
| Delaval | No | 33 | 2 | Yes |  | Before Analyzer | Visual | No | No |
| White Superior | Yes | 30 | 4 | Yes |  | Before Analyzer | Visual | Yes | ? |
| SAE (NDIR) |  |  |  |  |  |  |  |  |  |
| Waukesha | No | 33 | ~15 | Yes | Yes | Before Analyzer | Stripchart | No | -- |
| Cooper | No | 18-75 | <3 | Yes | Yes | Before Analyzer | Visual | No | -- |
| EMD (NDIR/NDUV) |  |  |  |  |  |  |  |  |  |
| GMC/EMD | No | 40 | 6 | Yes[a] | Yes[a] | Before Analyzer | Stripchart | Yes | -- |
| EPA (CL) |  |  |  |  |  |  |  |  |  |
| White/Alco | Yes | 60-80 | 6-8 | Yes |  | After Converter | Stripchart | Yes | Yes |
| Ingersoll-Rand | Yes | 60 | 6 | Yes |  | After Converter | Stripchart | Yes | Yes |

[a] EMD measures $NO_2$ with NDUV. No water traps are used.

## TABLE C-92. MANUFACTURER'S SAMPLING ERROR AS A PERCENT, RELATIVE TO EPA PROCEDURE

| Manufacturer's Practice | Sampling | | | Leak Check[d] | Recording (121) | NDIR[e] and Drier NDUV | Converter Check[f] |
|---|---|---|---|---|---|---|---|
| | Cold line, large residence time | Water trap before analyzer | | | | | |
| | | Cold line[b] | Hot line | | | | |
| DEMA | | | | | | | |
| Colt | -12[a] | -5 | | | | | |
| Cooper | – | -5 | | -25 | ±3 | | ↓ |
| Delaval | | -5 | | -25 | ±3 | | ↓ |
| White Superior | | | <-5[c] | | | | |
| SAE | | | | | | | |
| Waukesha | -12[a] | -5 | | -25 | | -40<br>+10 | |
| Cooper | | -5 | | -25 | ±3 | -40<br>+10 | |
| EMD | | | | | | | |
| GMC/EMD | | -5 | | | | -40<br>+20 | |

T-89

[a] Potential loss of NOx based on Data from Schaub and Beightol (118).

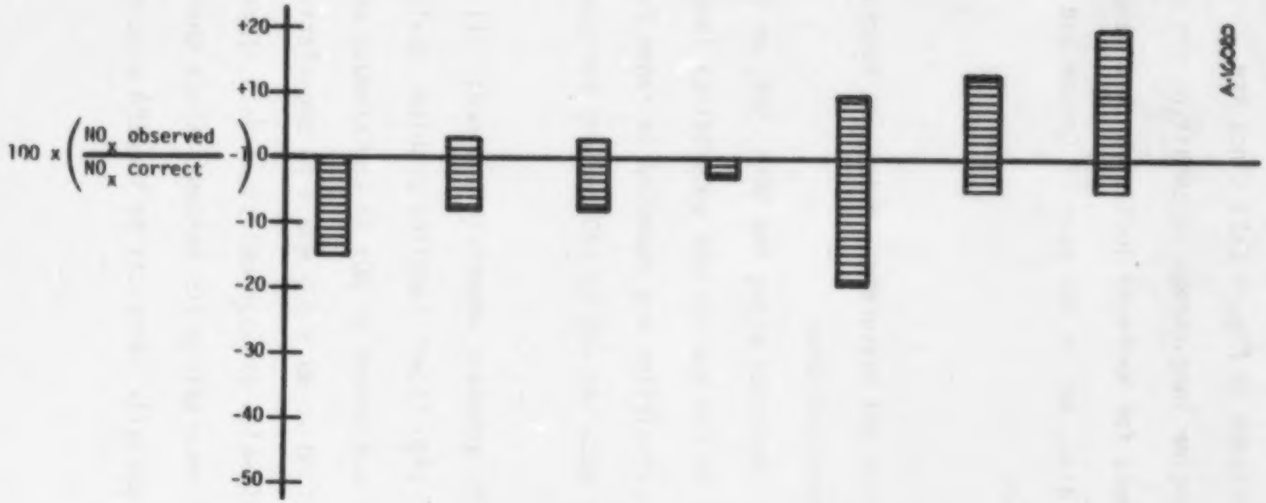
[b] Assumes that 5 percent of NO converted to $NO_2$ and lost in water trap. Based on data from Clemmens (119).

[c] Loss assumed to be less 5 percent since heated sample line used.

[d] Loss cannot be generalized. Twenty-five percent loss based on one example (120).

[e] Forty percent error for unconditioned driers, 10 percent for NDIR (122).

[f] Failure to perform converter checks could result in understated $NO_x$ values due to incomplete conversion of $NO_2$ to NO.



| Mfg | Colt | Cooper | Delaval | White Superior | Waukesha | Cooper | GMC/EMD |
|---|---|---|---|---|---|---|---|
| Practice | DEMA | | | | SAE | | EMD |

[a] Does not include potential error due to system leaks, and assumes chemical drier is conditioned.

Figure C-22. Measurement uncertainty relative to EPA procedure.[a]

these uncertainties substantially exceed the 4-percent scatter in $NO_x$ levels observed during verification tests of the EPA procedure.

NDIR and NDIR/NDUV instrument biases have been included in this figure and are based on the TRW study(123). Note that the potential negative 40-percent error of Table C-92 is not included in Figure C-22 since the manufacturers generally operate the engine long enough to condition the drier before measurements are taken. The data for Waukesha indicates that sampling errors can be as large as instrument bias, but in the case of Cooper and GMC/EMD, instrument biases predominate.

## Conclusions

The above analyses of instruments and measurement practices suggest the following conclusions about the reported data:

- The exhaust data which were reported using the DEMA, SAE, or EMD practices should be banded by the appropriate uncertainty level. In general, measurement uncertainties are expected to range from -15 to +5 percent for DEMA data and -20 to +20 percent for the SAE/EMD data.
- Waukesha's data contains the greatest uncertainty since: (1) only NO was measured, (2) other significant sampling problems could have been present, and (3) the amount of $NO_2$ in the exhaust was unknown; thus correction of NO to NO + $NO_2$ would be speculative.
- The Ingersoll-Rand and White-Alco exhaust emissions data appear to have the least measurement uncertainty (±5 percent) since their sampling procedure was essentially identical to the EPA procedure.

The previous analyses also lead one to conclude that the EPA heavy-duty engine method should be preferred over any other method previously used for sampling stationary reciprocating engines. Therefore, any test method developed for these engines should follow the same basic sampling procedures and utilize the same basic sampling system. In the mean time, the DEMA and SAE/EMD sampling practices should include more carefully specified sampling procedures, and the use of NDIR/NDUU instruments should be carefully reviewed because of their bias relative to CL instruments.

## C.4 THE EFFECT OF $NO_x$ CONTROL TECHNIQUES ON NONMETHANE HYDROCARBON EMISSIONS

It is well known that the application of $NO_x$ controls to IC engines can cause the emissions of other criteria pollutants to increase. Therefore, CO and HC emissions may require control as well as $NO_x$ levels. However, attempts to regulate HC emissions from large-bore engines are not straightforward since most data for these engines are in terms of total hydrocarbon (THC), rather than reactive HC which is the criteria pollutant.

Moreover, the Office of General Council (OGC) has indicated that EPA should establish, if feasible, standards for the criteria pollutant, NMHC, to avoid legal challenges of a THC standard under Section 111 of the Clean Air Act(124). The OGC added, however, that EPA could set standards on THC if such an approach: (1) simplifies sampling procedures, and (2) allows the use of existing data. If EPA follows this approach, they must show that a standard for THC would cost manufacturers and users no more than an standard for NMHC. The purpose of this memo is to investigate this issue.

Data from Section 4.4 indicated that, in general, the application of $NO_x$ controls to large-bore IC engines resulted in increases of 5 to 50 percent in THC levels (see Figure 4-52). However, the THC levels reported for the gas and dual-fuel engines contained both methane, ($CH_4$) a noncriteria pollutant, and NMHC, the regulated HC pollutant. (NMHC emissions are defined here as all other hydrocarbons in the exhaust sample.) If NMHC levels change differently than THC levels with the application of $NO_x$ controls, separate control regulations may be required for NMHC levels. Therefore, the data for THC and NMHC (where measured) from gas and dual-fuel engines are reexamined here to determine how NOx control techniques affect NMHC levels relative to THC levels.

In the following sections, uncontrolled levels of THC and NMHC will be compared for both new (laboratory) and installed (field engines). Then these emissions will be evaluated after the application of $NO_x$ control techniques that were reported in Section 4.4. In addition, the measurement techniques for both NMHC and THC will be briefly reviewed. Based on this evaluation, a recommendation will be presented concerning the separate control of NMHC emissions.

### C.4.1 Comparison of Controlled THC and NMHC Emissions from Gas and Dual-Fuel Engines

Previous investigators have shown that NMHC levels are generally less than 20 percent of the THC emissions from natural gas engines. Figure C-23 illustrates this conclusion using data reported by Southwest Research Institute (SwRI), and two engine manufacturers, Colt and Cooper. The SwRI data were from engines installed on gas transmission pipelines(125), whereas the manufacturer's data were from new engines that were measured in laboratories. All the THC data were measured with flame ionization detectors (FID's). Nonmethane emissions were determined as the difference in THC and methane levels. Cooper and Southwest measured methane emissions with a gas chromatograph. Colt physically removed all nonmethane portions of the sample and then measured the methane remaining with an FID.

Figure C-23 shows that THC levels for pipeline engines ranged from 1 to 9 g/hp-hr and were lower for four-cycle designs (1 to 3 g/hp-hr) compared to two-cycle levels (3 to 9 g/hp-hr). THC emissions from the engines tested by the two manufacturers were generally lower than those from the pipeline engines for both two- and four-cycle designs.

Although the THC levels for the new gas engines were lower than those for the installed engines, the NMHC levels were greater. As a consequence,

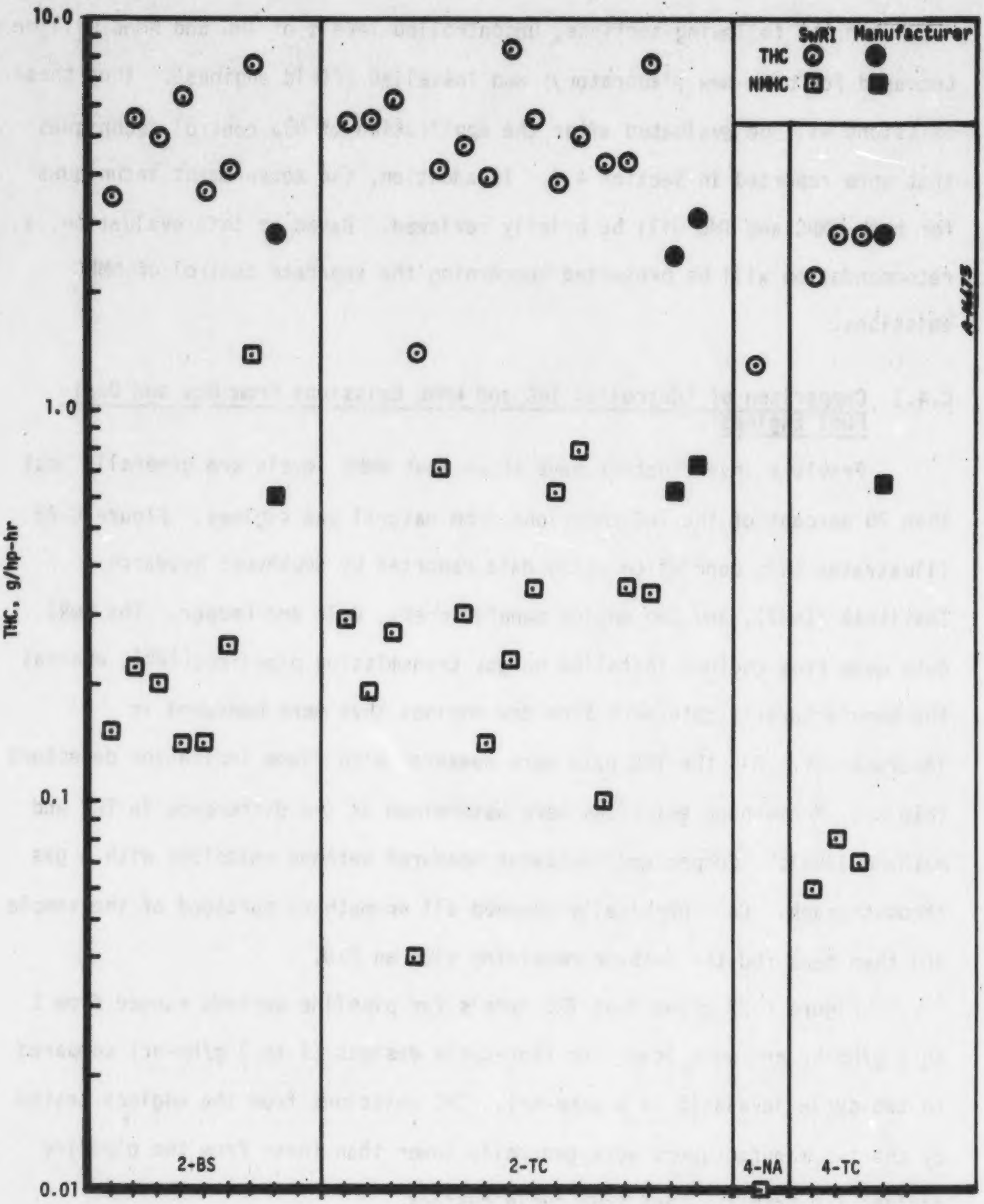


Figure C-23. Uncontrolled THC and NMHC levels from gas engines.

a greater fraction of the THC emission from the new engines were NMHC as compared to the pipeline units. On the average, NMHC emissions were only 6 percent of THC levels for the installed engines, but 23 percent for the new engines. These differences may be related to the condition of the installed engines which were tested as found. That is, older, installed gas engines may have had more blowby of unburned fuel and have been operating off peak combustion (i.e., requiring maintenance or an overhaul), thereby causing greater emissions of unburned fuel (methane). Furthermore, lower peak temperatures associated with less than optimum combustion may have resulted in fewer emissions of partially oxidized fuel (i.e., reactive hydrocarbons or NMHC).

Figure C-24 shows uncontrolled THC and NMHC levels for Colt and Cooper dual fuel engines. (No measurements of installed dual-fuel engines were available.) These data are similar to the limited data for manufacturer's gas engines, indicating that NMHC levels from these engines range from 10 to 20 percent of THC levels.

In the following sections, the levels of NMHC and the ratios of NMHC/THC for these gas and dual-fuel engines will be evaluated after the application of $NO_x$ controls. These comparisons will show whether NMHC levels respond differently to controls than to THC levels, and, therefore, whether any standards would have to be written in terms of NMHC.

### C.4.2 The Effect of $NO_x$ Control Techniques on THC and NMHC Levels

Figures C-25 through C-30 illustrate how the application of $NO_x$ control techniques affect THC and NMHC levels from gas and dual-fuel engines. Note that data were reported for both pipeline engines and those tested in manufacturers' laboratories. In general, these figures show that derate

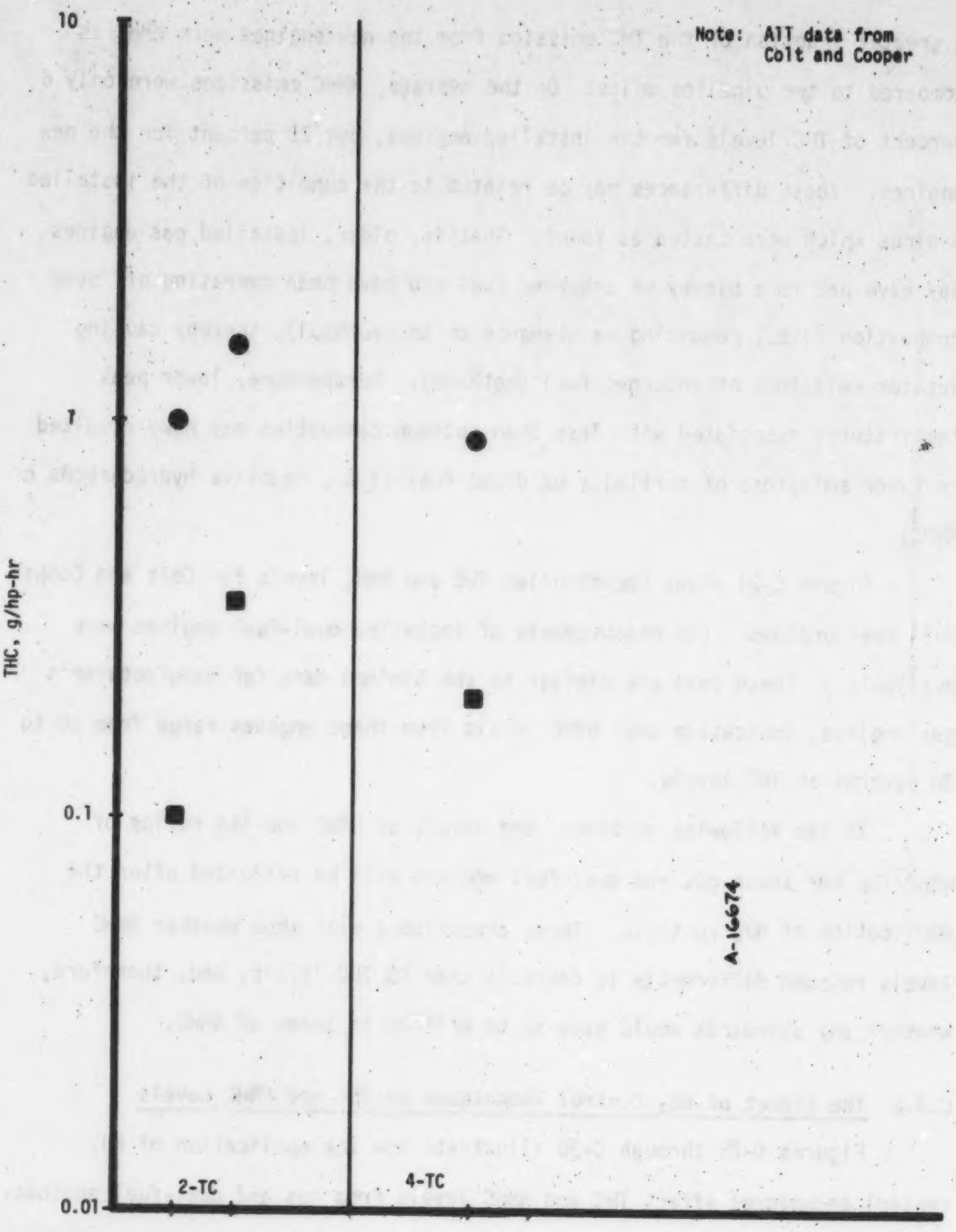


Figure C-24. Uncontrolled THC and NMHC levels for dual fuel engines.

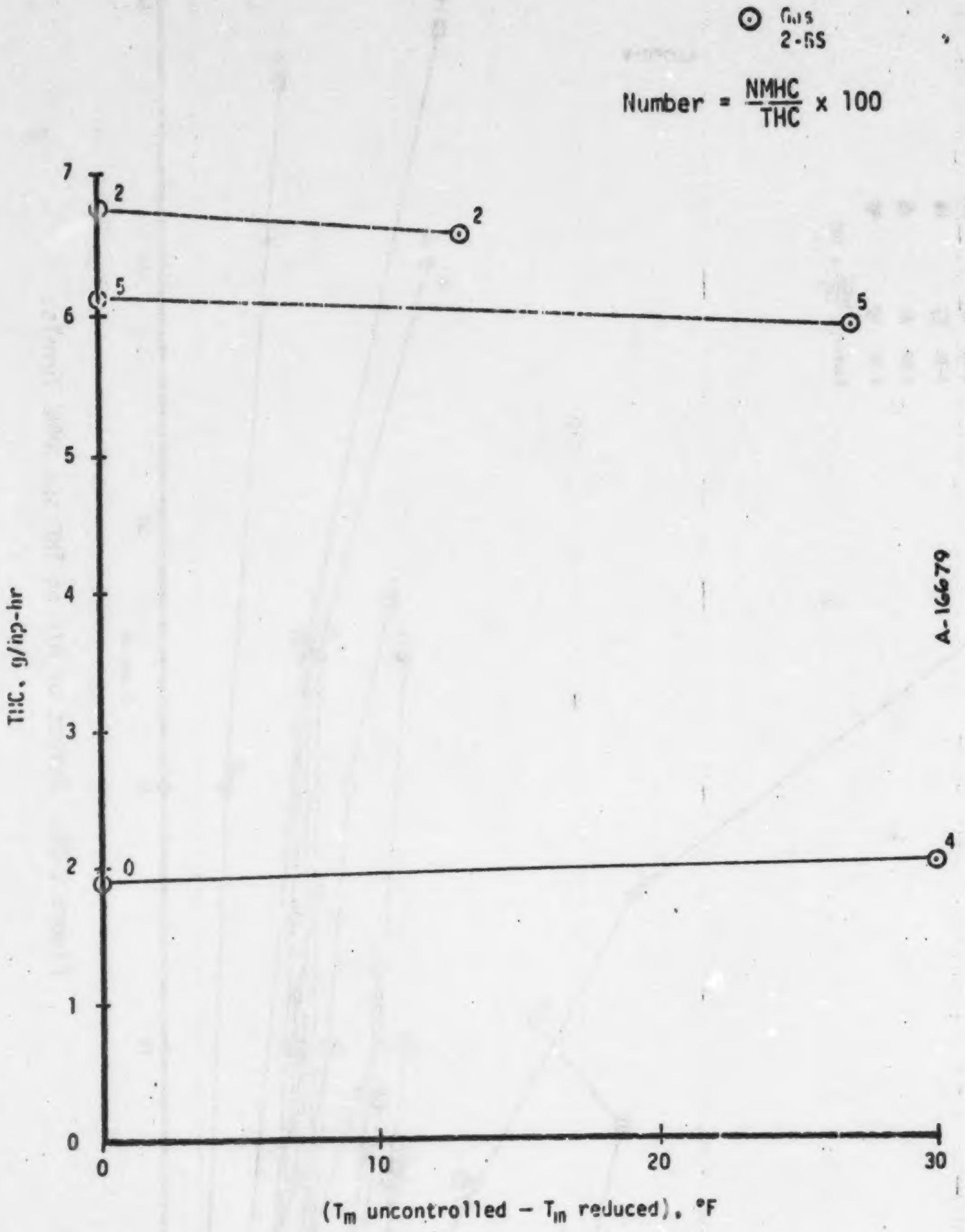


Figure C-25. Effect of derating on THC and NMHC levels.

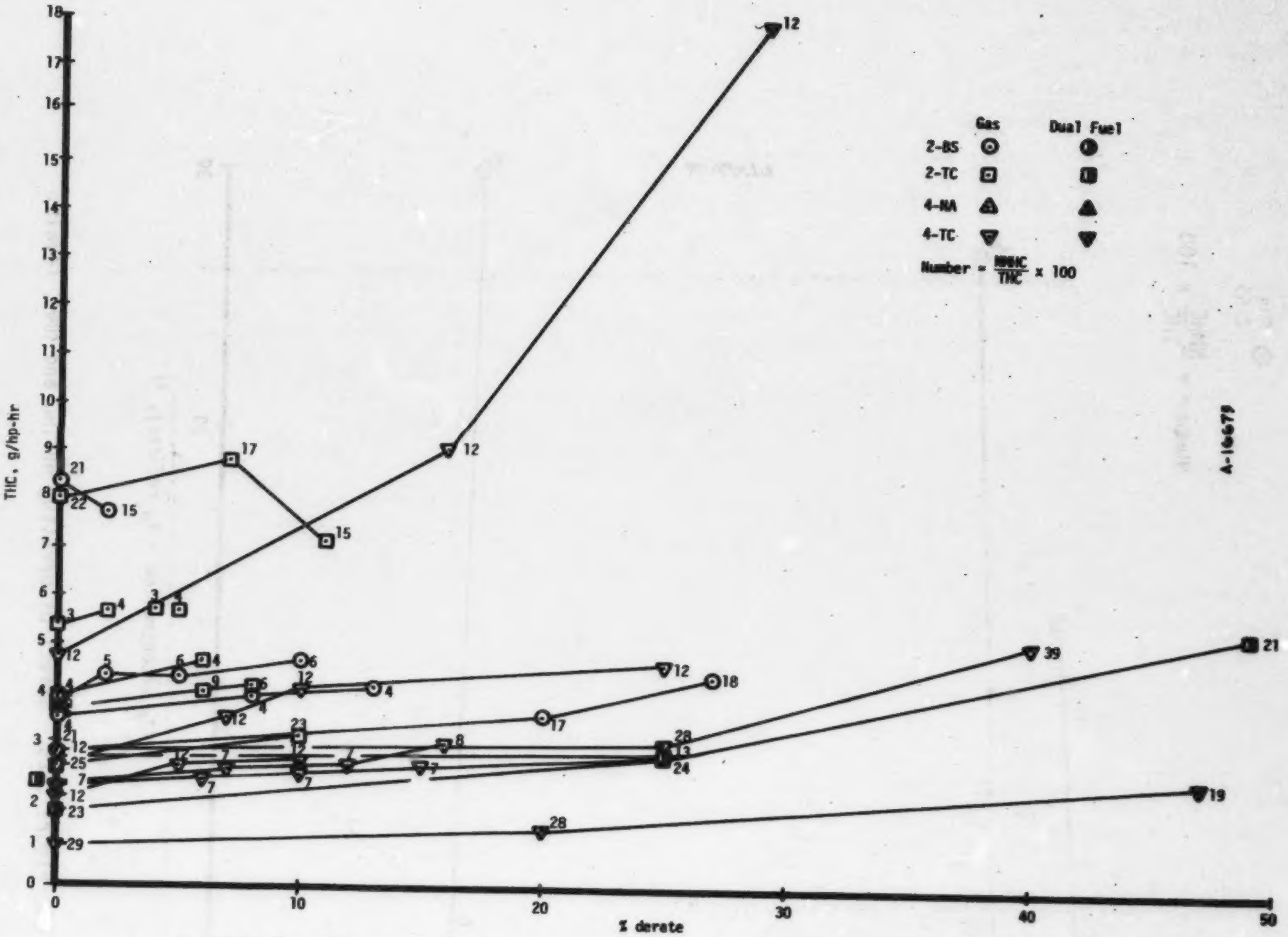


Figure C-26. Effect of A/F on THC and NMHC levels.

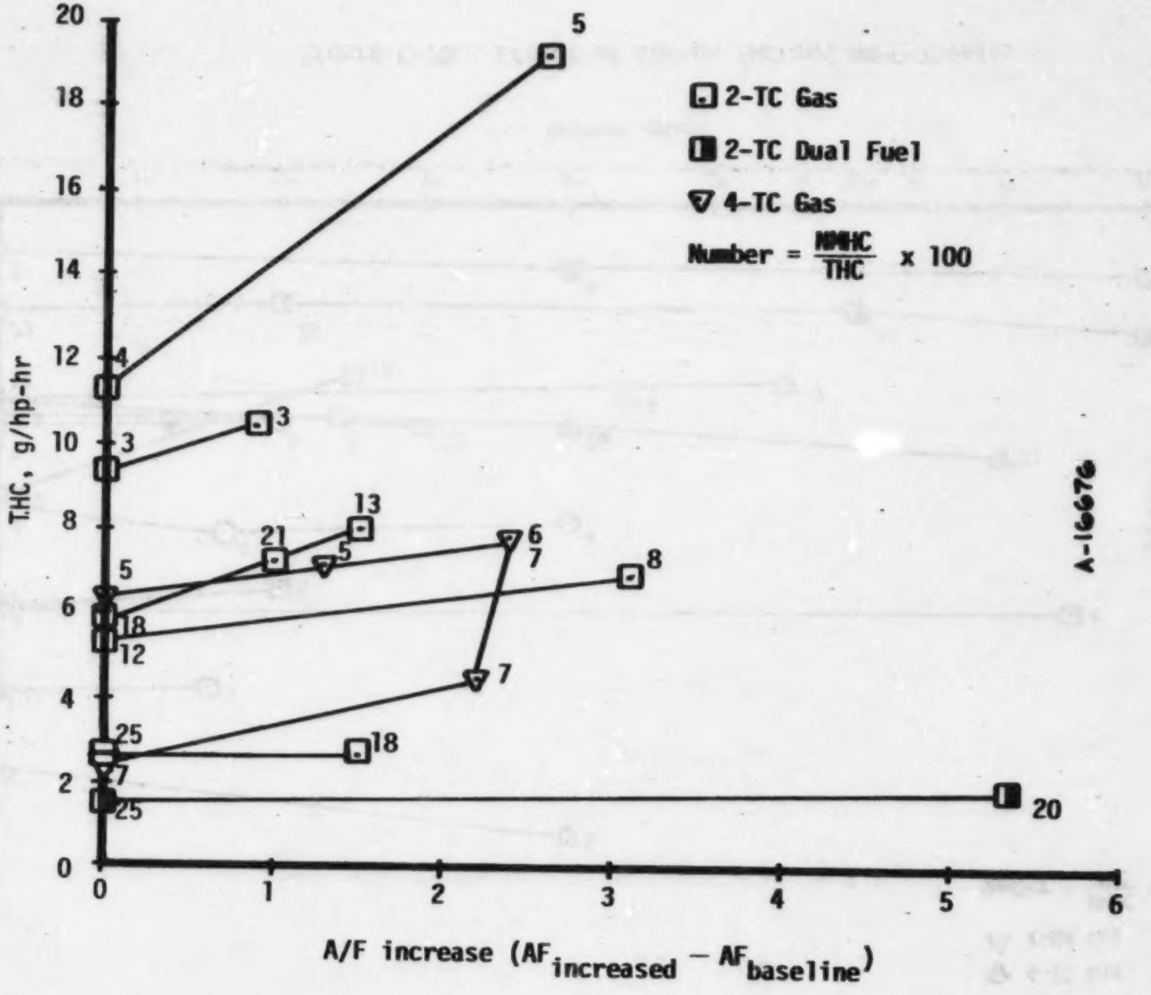


Figure C-27. Effect of retard on THC and NMHC levels.

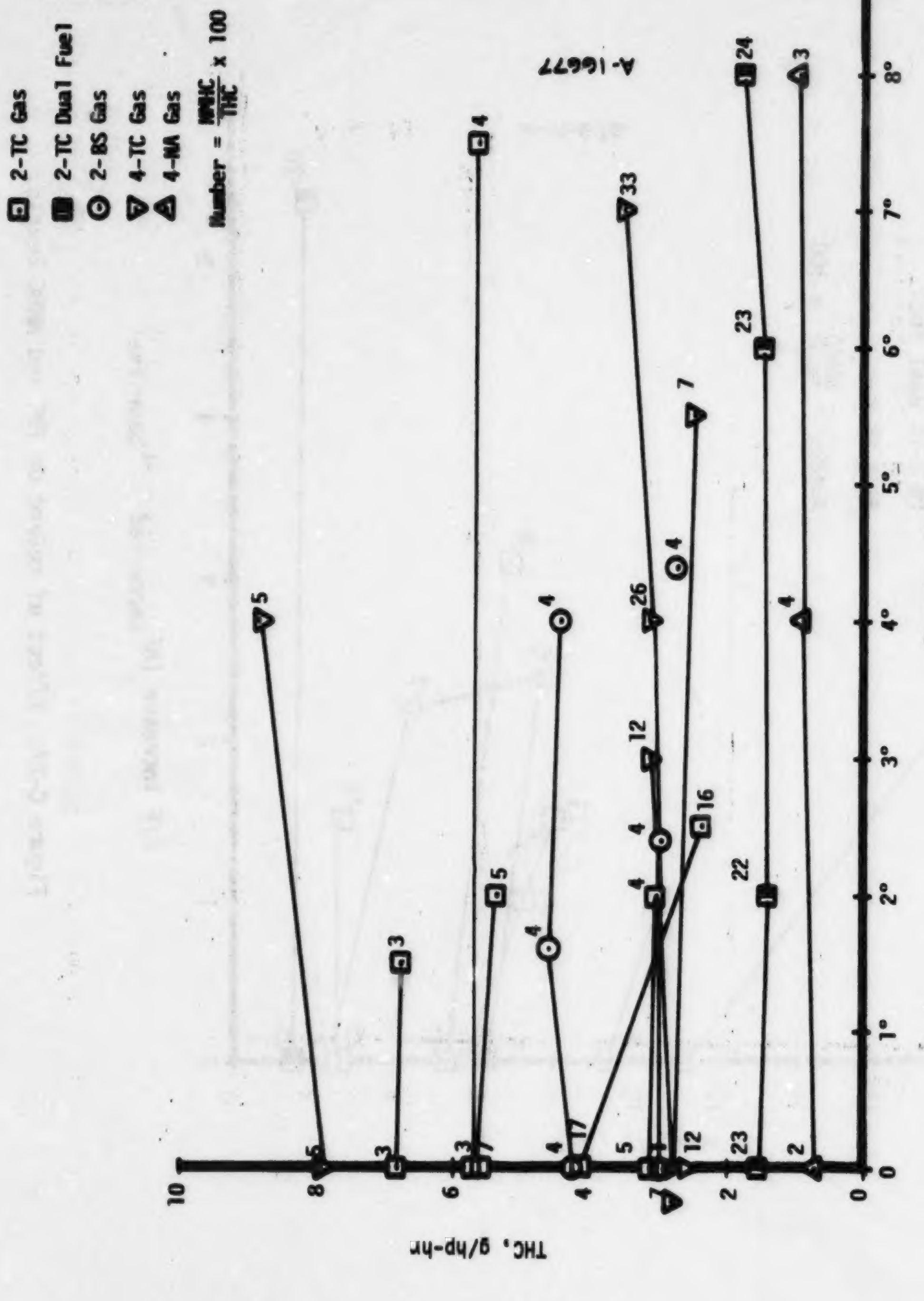


Figure C-28. Effect of EGR on THC and NMHC levels.

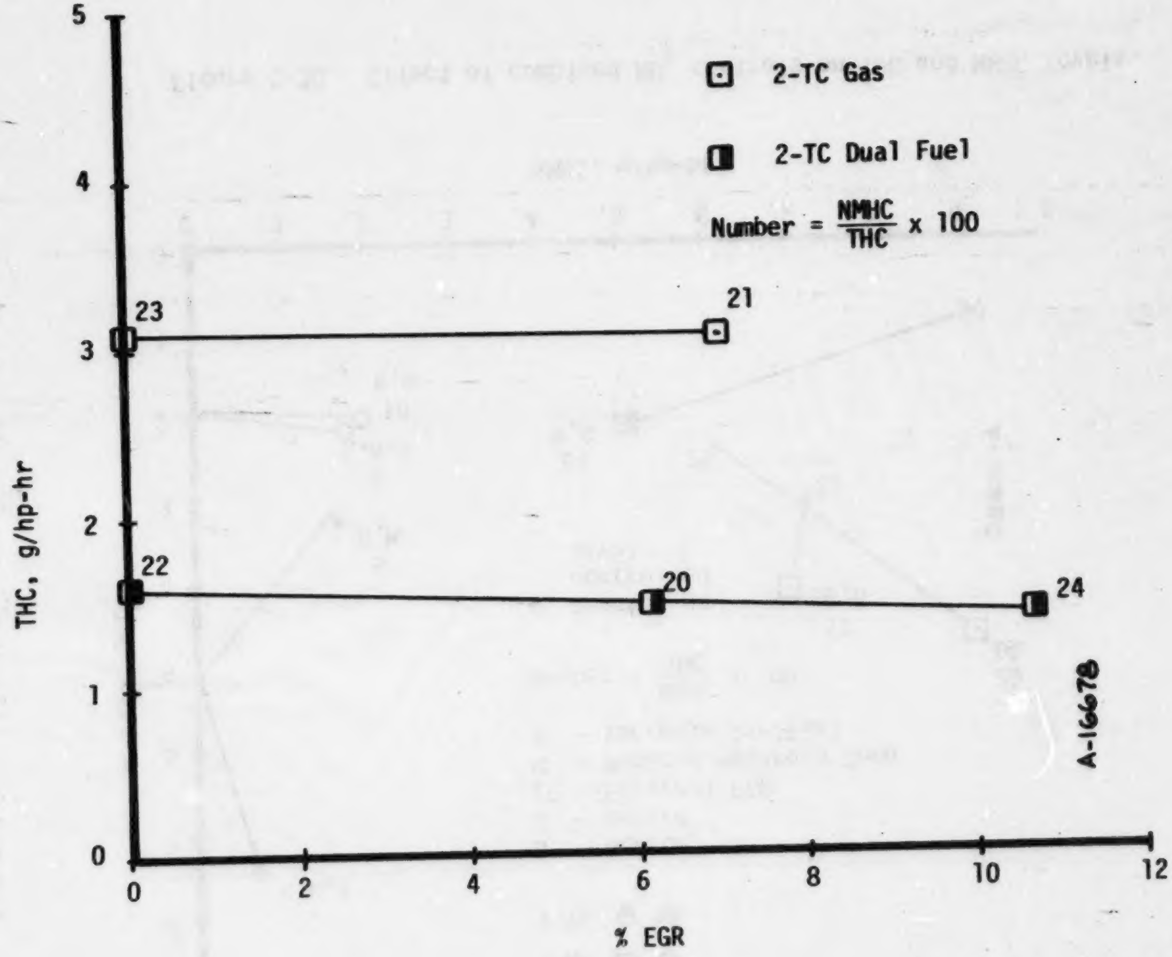


Figure C-29. Effect of manifold air cooling on THC and NMHC levels.

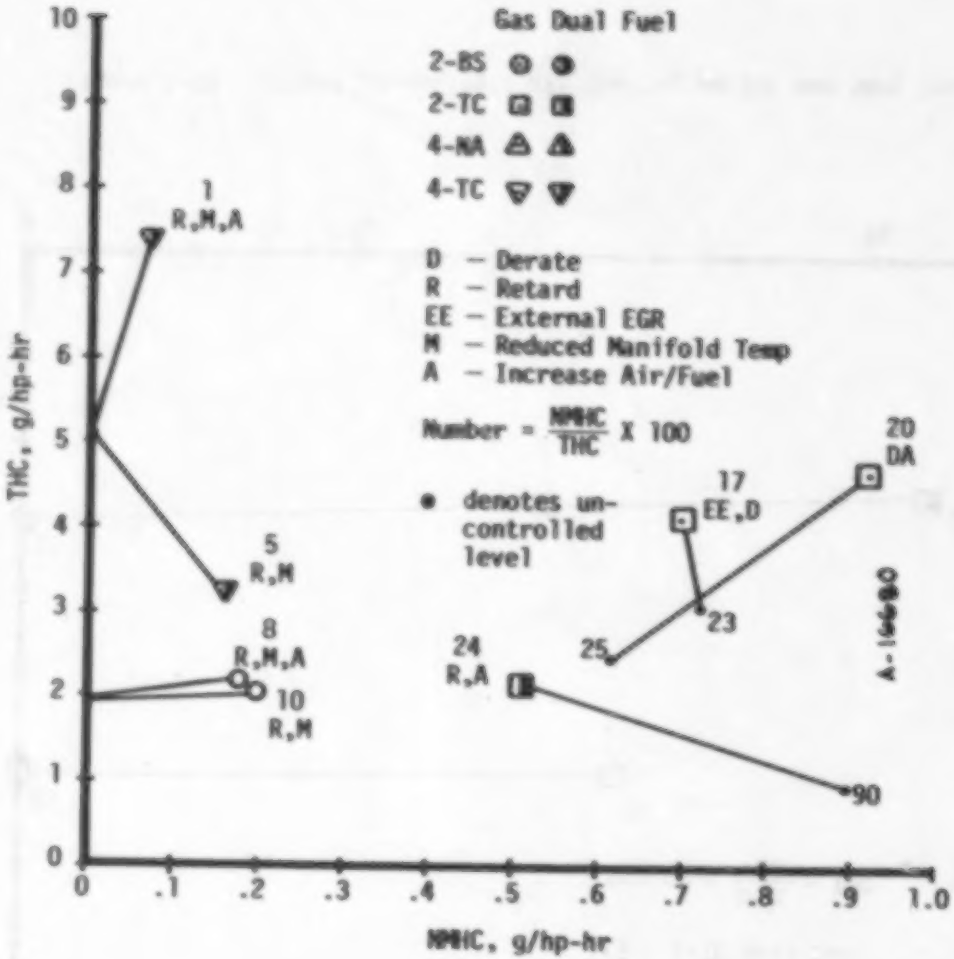


Figure C-30. Effect of combined $NO_x$ controls on THC and NMHC levels.

and air-to-fuel ratio changes cause THC emissions to increase, but retard (R), external EGR, and manifold air temperature (MAT) reductions do not change the uncontrolled THC. Furthermore, for all of these techniques, NMHC levels remained about the same percentage of THC as they were in the uncontrolled engines. That is, NMHC emissions from both gas and dual-fuel engines change in the same proportion as do THC levels, or remain in the same proportion (where THC remained unchanged) with the application of $NO_x$ controls.

Figure C-25 shows that, in general, derating caused THC to increase less than 50 percent, although extreme amounts of derate (>25 percent) caused uncontrolled THC levels to increase from 1 to 3 g/hp-hr to 2 to 5 g/hp-hr. In one case, THC emissions increased from 5 to 18 g/hp-hr for an installed pipeline engine that, possibly, required maintenance. In all cases, NMHC levels ranged from 5 to 25 percent of THC levels and did not change with derating. The increases in THC levels are probably related to the lower exhaust and peak temperatures of part load operation.

Similarly, Figure C-26 shows that increases in A/F cause moderate (10- to 25-percent) increases in THC levels. One 2-TC installed gas engine was the exception; THC levels increased 75 percent from 11 to 19 g/hp-hr, and may be related to a worn condition of the engine requiring maintenance (i.e., excessive blowby). Increases in A/F of gas engines that are already operating lean tend to lower peak combustion temperatures and prevent complete combustion. Consequently, HC emissions increase, although not as rapidly as with decreases in A/F from rich (less than stoichiometric) operation (see Figure 4-29). Note that the ratio of NMHC to THC remains essentially constant with increases in A/F.

Figures C-27 through C-29 indicate that retard, EGR, and MAT do not cause THC or NMHC levels to increase. The application of retard delays the combustion of the air/fuel mixture, resulting in higher exhaust temperatures which, combined with an excess of oxygen, effectively oxidizes most remaining unburned fuel. The application of this technique, however, causes fuel consumption to increase (see Figure 4-27).

In contrast to retard, EGR and MAT controls usually cause no increase in fuel consumption. It follows that if complete combustion is maintained (evidenced by unchanged fuel consumption) HC emissions should remain low or unchanged. With the application of EGR, unburned HC are recirculated or trapped and then combusted during the next cylinder firing. The limited data for EGR, illustrated in Figure C-28, show that NMHC and THC levels remain unchanged.

With the reduction of manifold air temperature, however, HC emissions can be expected to increase as the reaction of this specie proceeds at a lower temperature. Data shown in Figure 4-52 indicate that THC levels increase, but less than 25 percent, with manifold temperature reductions. Nevertheless, the limited data for both NMHC and THC levels shown in Figure C-29 indicate that these emissions remain unchanged.

Figure C-30 shows how the application of combinations of controls affects NMHC and THC emissions. The results of this figure are mixed. It should be noted that this data is limited, representing six engines from two manufacturers. For three of the six engines shown, combined controls cause THC and NMHC emission to increase in the same proportion. However, data for two 2-TC designs (one dual fuel, the other gas) indicated that NMHC emissions decreased (when THC levels increased) with the applications of: (1) retard and air-to-fuel for the dual fuel unit, and (2) EGR and derate for the gas engine.

Opposing this result and of more concern, one 4-TC design showed that NMHC emissions increased from 0 to 0.2 g/hp-hr when THC emissions decreased from 5 to 3 g/hp-hr with the application of retard and MAT. Nevertheless, the NMHC level is small, representing only 5 percent of the THC level. It should also be noted that, despite precautions to avoid sample degradation, the manufacturer of this engine experienced some inconsistencies in determining NMHC levels (i.e., measured methane levels exceeded THC levels). Therefore, the following section will briefly consider the procedures for measuring both NMHC and THC before a recommendation concerning the separate control of NMHC emissions is presented in Section C.4.4.

### C.4.3 Measurement Practices for NMHC and THC Emissions

The development of a separate standard for NMHC emissions is largely dependent on the demonstration of techniques for detecting NMHC directly, or those which detect NMHC emissions indirectly by measuring methane and THC. The NMHC level is then determined by the difference of the two measured levels. It should be noted that the accuracy of indirect techniques depends on the accurate determination of both methane and THC levels.

At the present time, there are basically three methods available for determining NMHC emissions. These are(126):

- Gas Chromatographs (GC): Provide direct measurement of methane by separation from other hydrocarbons. NMHC measurement is obtained indirectly by subtraction of methane from THC or directly by measurement of remaining nonmethane hydrocarbons. When NMHC is measured directly, the methane value is also obtained and can be subtracted from a THC measurement (with FID) as a check of the NMHC measurement.

- Nondispersive Infrared (NDIR): Also provides direct measurement of methane. NMHC measurement is obtained indirectly by subtraction of methane measurement from independent THC measurement.
- Selective Combustor (SC): Gives a direct measurement of methane by selective combustion of all nonmethane hydrocarbon, allowing measurement of the remaining methane on a conventional THC analyzer such as a flame ionization detector (FID). NMHC measurement is obtained indirectly by subtraction of methane from THC.

All of these NMHC techniques use a conventional heated flame ionization detector (FID) to make one measurement of either NMHC using a GC, or THC using a GC, NDIR, or SC.

Gas chromatographs are commercially available and were used by Cooper(127) and SwRI to report NMHC levels from their engines. This method, however, cannot perform real-time analysis (continuous rather than bag or batch sampling), a capability desired by development engineers and present in other major emission instruments (e.g., $NO_x$, CO, $SO_x$ analyzers). In addition, this instrument requires specialized training to operate and maintain.

Selective combustors have only recently become available commercially. They are simple and inexpensive compared to a GC but apparently require more development. This is because not all lighter paraffins (e.g., ethane and propane) are eliminated from the test sample during the selective removal of all NMHC emissions. This results in higher methane (or smaller NMHC) levels than were actually present. Colt used this method to report NMHC levels from their engines. However, no comparisons with GC data were available to determine if this effect may have caused the Colt NMHC data to be understated.

The third NMHC instrument, the NDIR, exists as a prototype and also requires more development before becoming available commercially. Nevertheless, the instrument has demonstrated accuracy and freedom from interference for either batch or continuous sampling of methane emissions. It is also believed that it will require greater maintenance than existing NDIR's since it is more complex.

It should be noted that Fisher and Goodwin(128) supported the development of standards for NMHC's. They noted, however, that although measurements of NMHC levels were feasible, significant time and expense would be required to adequately develop the necessary instruments and procedures.

Cooperative studies of the SAE measurement method have also indicated that futher development of heated FID's is required to reduce measurement uncertainties(129). The most recent study consisted of a comparison of data from one diesel engine emission source measured during round-robin tests. The results of this study showed that THC levels varied by 10 percent (on the average) for a given manufacturer and 22 percent among manufacturers. It was concluded that further development of this measurement method was necessary to reduce this scatter.

The preceding discussion indicates that measurement methods for NMHC have not been adequately demonstrated at this time. In addition, studies of FID instruments indicate that considerable measurement uncertainty can arise from these instruments even though an accepted practice is followed.

### C.4.4 Recommendations for the Control of NMHC

Based on the preceding results, it is recommended that any control of NMHC levels be based on the measurement of THC levels. This can be supported by recognizing that: (1) in general, NMHC levels change in the same manner as THC levels with the application of $NO_x$ control techniques; and (2) monitoring NMHC levels accurately is considerably more difficult than for THC since gas chromatographs or physical conditioning of the sample is required. Therefore, if standards for hydrocarbons are deemed necessary, they should be based on the reduction of THC, rather than NMHC, levels.

## C.5 DEVELOPMENT OF CONVERSION $BSNO_x$ TO $NO_x$ (ppm) CORRECTED TO 15 PERCENT $O_2$

All emissions data in Alpendix C.1 had been reported in terms of brake specific emissions, $BSNO_x$ (g/hp-hr). In addition, applying percent emissions reduction to the sales weighted uncontrolled emissions levels (section 4.3.4) results in emission limits having the brake-specific $NO_x$ format. Since a concentration format (ppm) has been selected for the standard, a method must be derived to convert the $BSNO_x$ limits to concentration limits.

### C.5.1 Converting $BSNO_x$ to $NO_x$ Concentration Format

By definition, the volumetric concentration of $NO_x$ in the exhaust is simply the volume of $NO_x$ divided by the total exhaust volume. If the mass concentration is known, dividing each (the mass of $NO_x$ and exhaust) by the respective molecular weights will give the volumetric concentration. This may be equally done on a brake-specific basis. Therefore, since the molecular weight of $NO_x$ will be defined equal to $NO_2$ ( 46), the volumetric concentration of $NO_x$ can be written as

$$NO_x \text{ (ppm)}_{x\text{concentration}} = \frac{\dfrac{BSNO_x}{46}}{\dfrac{\text{Brake specific exhaust gas, g/hp-hr}}{\text{molecular weight of exhaust gas}}} \qquad (1)$$

Both brake-specific $NO_x$ limits and the molecular weight of $NO_x$ are known. However, the brake specific exhaust gas flow and the modecular weight of the exhaust gas are in general not available. Furthermore, to set emission limits for each fuel type, the average ratio of exhaust flow to exhaust gas molecular weight must be determined. This may be done by assuming properties of the fuel and a reference 15 percent $O_2$ concentration in the exhaust gas. The following procedure has been usedto determine the ratio of exhaust gas flow to exhaust gas modecular weight, using data obtain from available brake-specific fuel consumption.

For any hydrocarbon fuel, the chemical reaction may be written as:

$$C_xH_y + Z\,O_2 + 3.76Z\,N_2 \rightleftharpoons x\,CO_2 + y/2\,H_2O + (Z - x - y/4)\,O_2 + 3.76\,N_2 \qquad (2)$$

This is a generalized equation, ignoring the small concentrations of other constituents such as $NO_x$, HC, CO etc., which are less than 1.0 percent. The moles of exhaust gas produced per mole of fuel burned is therefore $[x + y/2 + (Z - x - y/4) + 3.76\,Z]$; and for dry products water is removed leaving $[x + (Z - x - y/4) + 3.76\,Z]$ moles of dry exhaust. This reduces to $[4.76\,Z - y/4]$ total moles of exhaust gas. The molecular weight of the gas is therefore the ratio of each constituent multiplied by the respective constituent molecular weight:

$$M_{\text{exhaust gas}} = \frac{x}{[4.76Z - y/4]}\,44 + \frac{(Z - x - y/4)}{[4.76Z - y/4]}\,32 + \frac{3.76Z}{[4.76Z - y/4]}\,28 \qquad (3)$$

or

$$M_{\text{exhaust gas}} = \frac{1}{[4.76Z - y/4]}\,(44x + 32Z - 32x - 8y + 105.28Z) \qquad (3a)$$

reducing to

$$M_{\text{exhaust gas}} = \frac{1}{[4.76Z - y/4]} (12x - 8y + 137.28Z) \qquad (4)$$

Since the standard assumes a 15 percent oxygen concentration in the dry exhaust gas, the moles of air, (Z), can be determined in terms of carbon and hydrogen in the fuel, (x and y), as follows:

$$0.15 = \frac{Z - x - y/4}{4.76Z - y/4} \qquad (5)$$

$$0.714Z - 0.0375y = Z - x - 0.25y \qquad (5a)$$

$$Z = \frac{x + 0.2125y}{0.286} \qquad (5b)$$

$$Z = 3.5x + 0.743y \qquad (6)$$

Therefore, by substitution into Equation (4),

$$M_{\text{exhaust gas corrected, 15\% } O_2} = \frac{492x + 94y}{16.64x + 3.29y} \qquad (7)$$

The exhaust mass flow must be determined next. Since the total mass flow remains constant, the mass of exhaust gases per mole of fuel burned is:

$$\dot{m}\text{ dry exhaust} = \dot{m}\text{ fuel} + \dot{m}\text{ air} - \dot{m}\text{ water formed} \qquad (8)$$

From equation (1) above, (Z) moles of air are burned with each mole of fuel, and (y/2) moles of water are formed. By multiplying by the molecular weight of each, the mass flows are as follows:

$$1 \times (12x + y) + Z[32 + 3.76(28)] - (y/2)(18) \qquad (9)$$

Substituting equation (5) again (assuming 15 percent oxygen in exhaust);

$$\frac{\dot{m}\text{ dry exhaust}}{\text{mole fuel}} = (12x + y) + (3.5x + 0.743y)(32 + 3.76(28)) - (y/2)(18) \qquad (10)$$

$$\frac{\dot{m}\text{ dry exhaust}}{\text{mole fuel}} = 492x + 94y \qquad (11)$$

Dividing by the molecular weight of the fuel yields the mass of exhaust gases per mass of fuel, assuming 15 percent $O_2$ in the exhaust stream

$$\frac{\dot{m}\ \text{dry exhaust}}{\dot{m}\ \text{fuel}} = \frac{492x + 94y}{12x + y} \tag{12}$$

By multiplying this value by the brake-specific fuel consumption (BSFC) the brake-specific exhaust flow is determined.

$$\dot{m}\ \text{dry exhaust} = \text{BSFC}\left(\frac{492x + 94y}{12x + y}\right) \tag{13}$$

The ratio of exhaust mass flow to exhaust gas molecular weight can now be determined. From equation (13) and (7):

$$\frac{\text{Brake specific exhaust flow}}{\text{Molecular wt. of exhaust gas}} = \frac{\text{BSFC} \times \left(\frac{492x + 94y}{12x + y}\right)}{\left(\frac{492x + 94y}{16.64x + 3.29y}\right)} \tag{14}$$

This reduces to

$$\text{BSFC} \times \left(\frac{16.64x + 3.29y}{12x + y}\right) \tag{15}$$

Substitution into equation (1) and converting to parts per million (ppm) gives:

$$NO_{x_{15}} = \frac{BSNO_x \times 10^6}{(46)\ (BSFC)\left(\frac{16.64x + 3.29y}{12x + y}\right)} \qquad (16)$$

where $NO_{x15}$ is corrected to 15 percent $O_2$ in the exhaust.

To employ this equation, the values of x and y for the fuel must be known. In addition, BSFC must have the same units as $BSNO_x$ for substitution into equation (16). Thus, the following data for the fuel is required: 1) actual carbon and hydrogen makeup of the fuel $C_xH_y$ or the hydrogen/carbon ratio y/x; and 2) the fuel lower heating value Btu/g, since most fuel consumptions are expressed in Btu/hp-hr.

### C.5.2 Determining Emission Limits

The above conversion method was used to establish emission limits in the following manner. First, equation (16) and the data in Appendix C.1 were used to generate plots of $BSNO_x$ and $NO_{x15}$ for each fuel type. These are shown in Figures C-31, C-32 and C-33. For diesel and dual-fuel engines, diesel fuel with a hydrogen/carbon ratio of 1.8 and a LHV of 18,320 Btu/lbm was assumed. Natural gas was assumed to have a LHV of 20,000 Btu/lbm, and a hydrogen/carbon ratio of 3.5. The best fit line was then determined through all points as shown on the figures.

Once these best-fit lines were established, average BSFC corresponding to the lines were determined. These equal 7057 Btu/hp-hr for diesel, 7163 Btu/hp-hr for dual fuel, and 7609 Btu/hp-hr for gas engines. By averaging these values, an average BSFC of 7276 Btu/hp-hr was achieved, which represents an engine efficiency of 35 percent for all engines. This average value was placed into equation (16) and by using

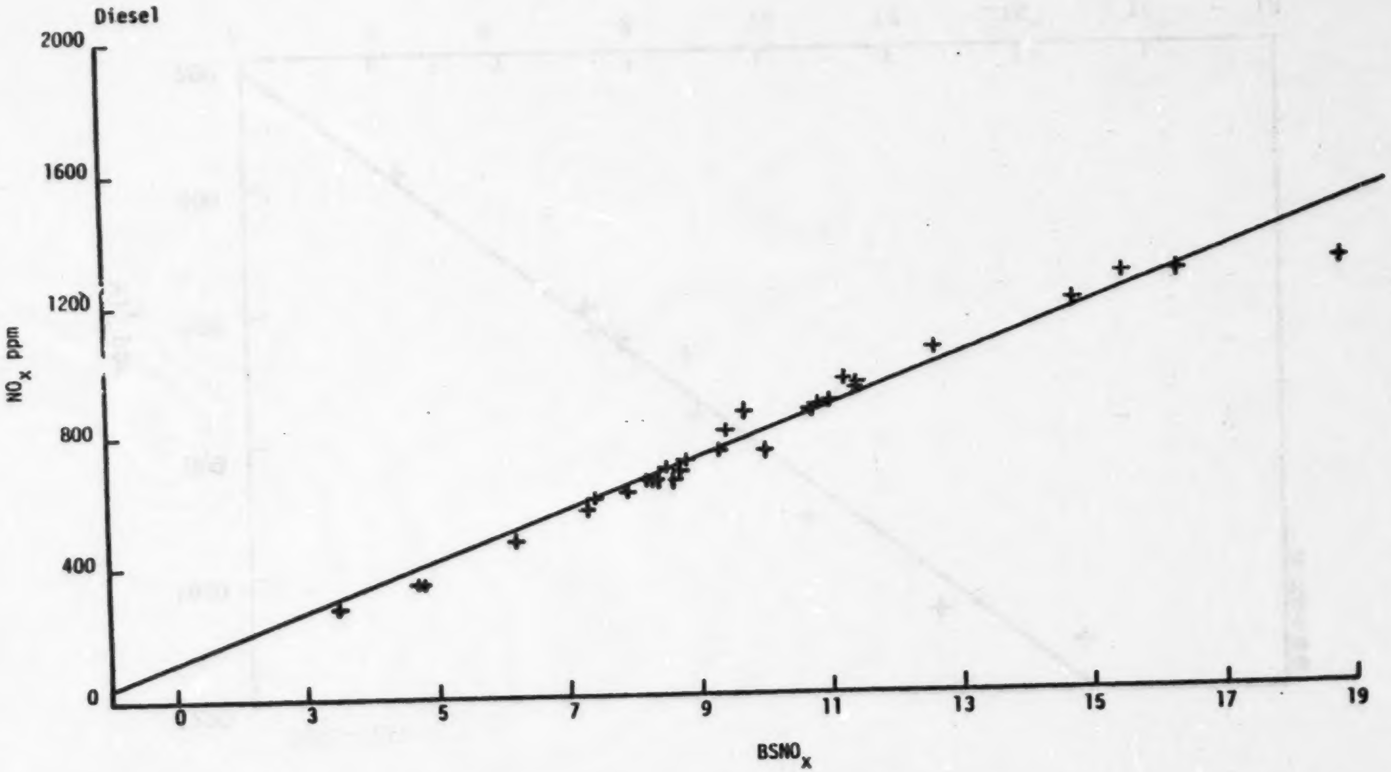


Figure C-31. BSNO$_x$ converted to NO$_x$ ppm; from estisting uncontrolled diesel engine data.

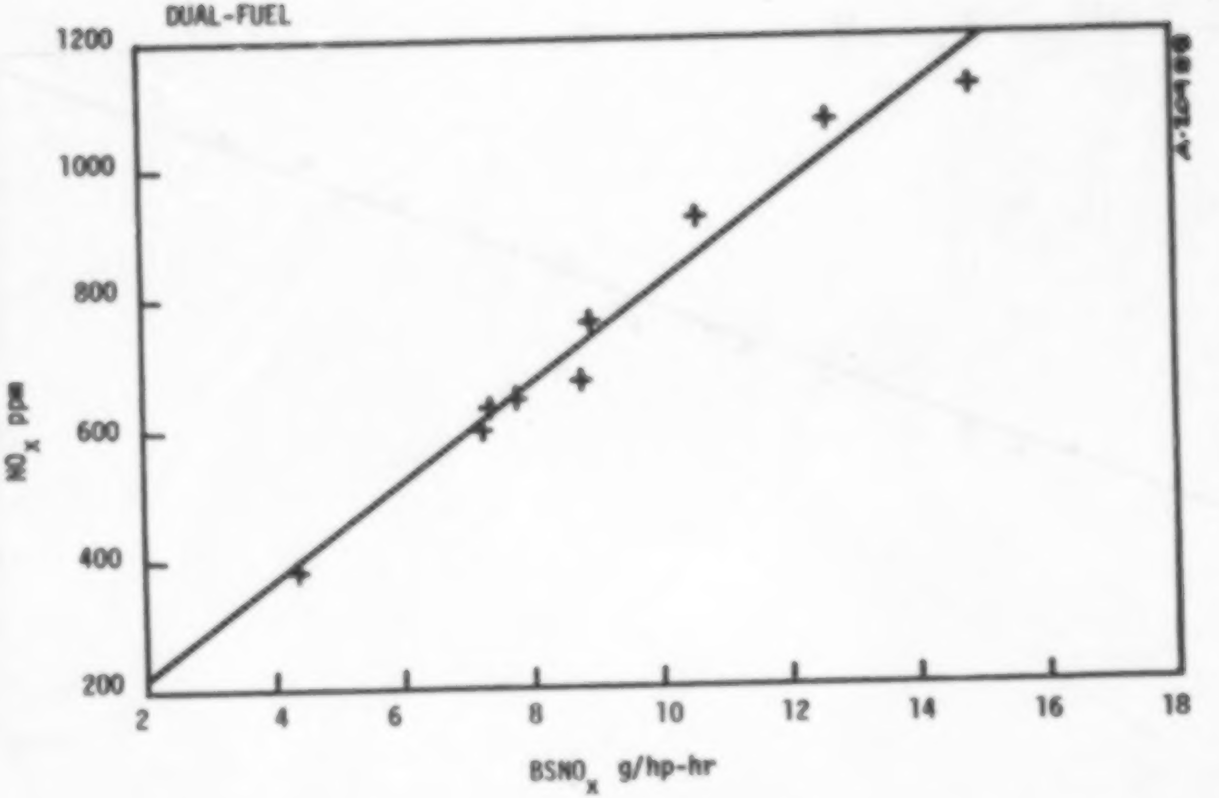


Figure C-32. $BSNO_x$ converted to $NO_x$ ppm; from existing uncontrolled dual-fuel engine data, assuming the engine being operated on 100-percent diesel fuel.

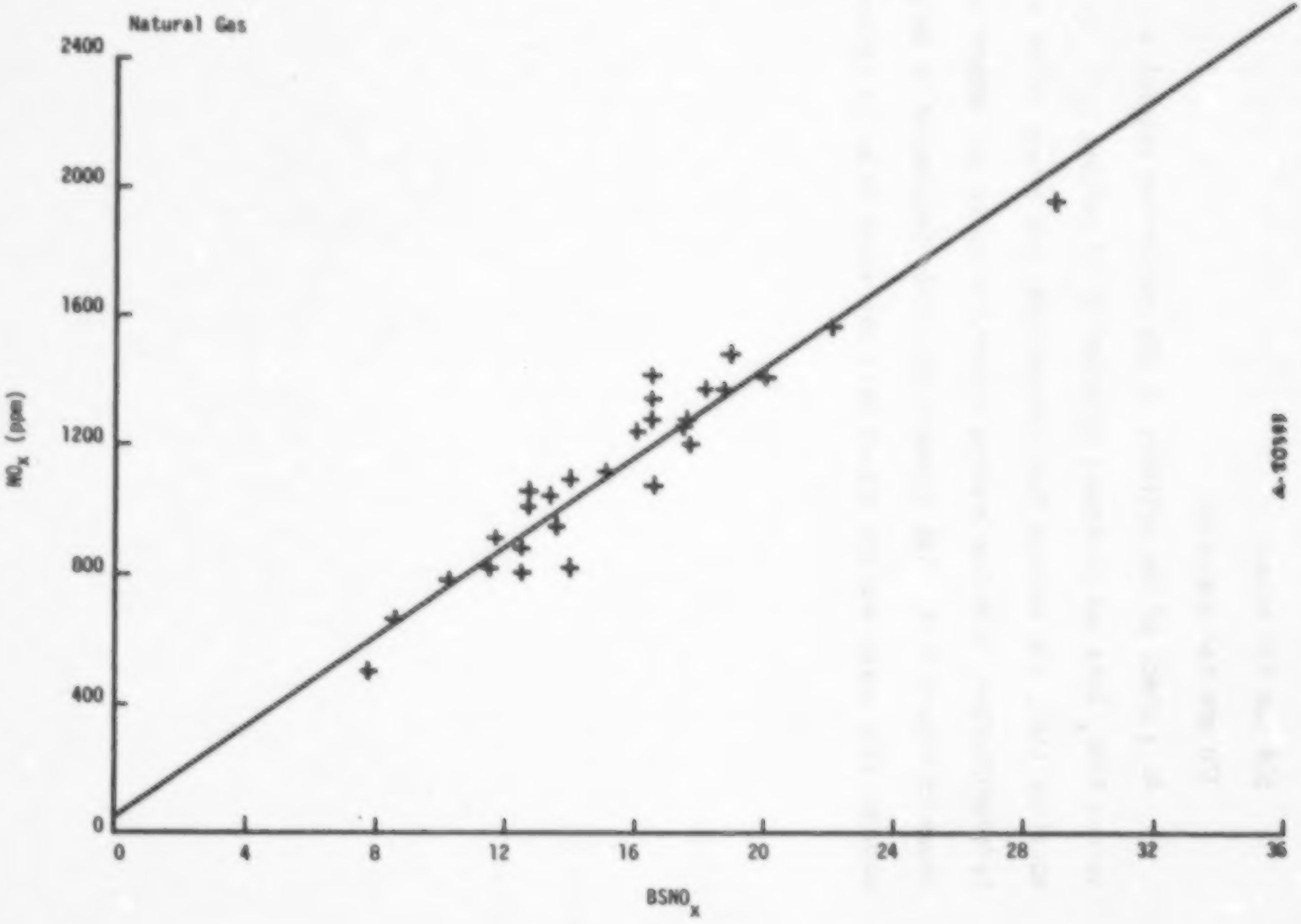


Figure C-33. $BSNO_x$ converted to $NO_x$ ppm; from uncontrolled natural gas engine data.

the limits of 5 g/hp-hr for dual fuel, 9 g/hp-hr for gas, and 7 g/hp-hr for diesel, (and the above fuel properties) the following $NO_x$ emission limits were determined:

654 ppm for gas

518 ppm for diesel

370 ppm for dual-fuel

As a check of the validity of the conversion method, all raw ppm versus $BSNO_x$ data was plotted (corrected to 15 percent $O_2$). By using equation (16), the average fuel consumption, and a mean value of the hydrogen/carbon ratio the average conversion curve was determined and is shown in Figure C-34. The standard deviation (expressed in percent) between this curve and the actual data was found to be 10 percent.

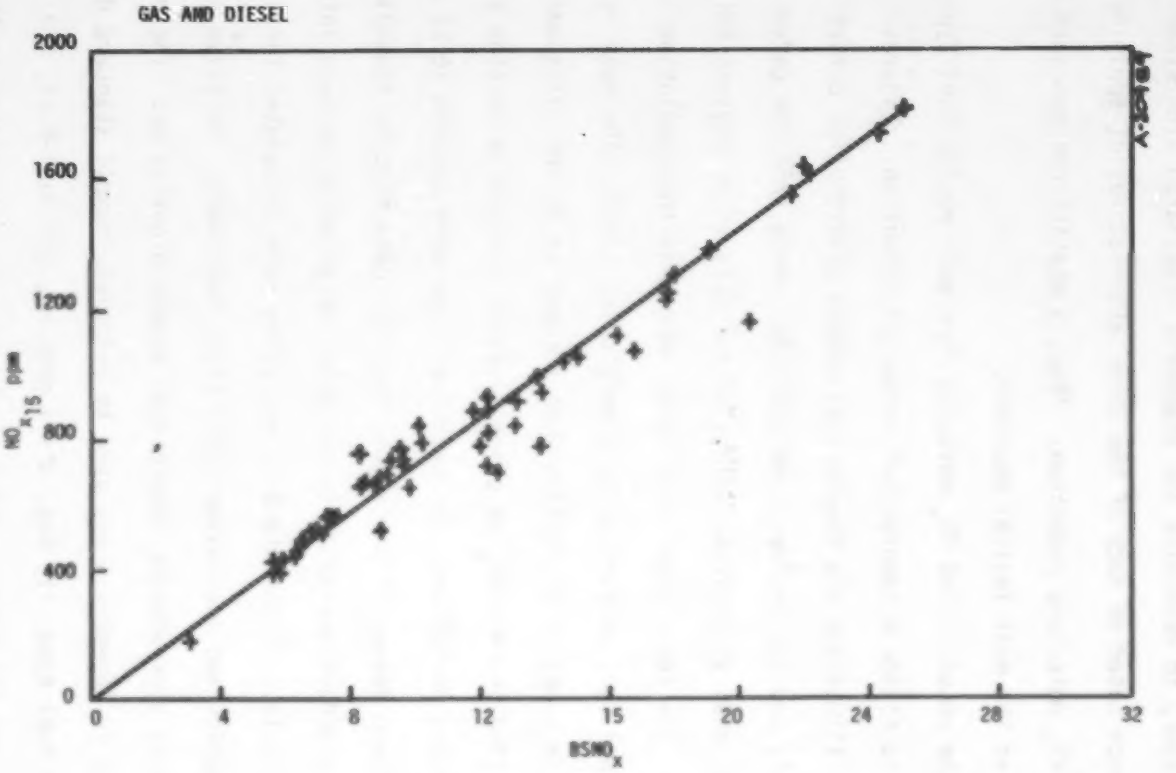


Figure C-34. Equation (16) plotted against raw diesel and gas ppm and $BSNO_x$ data.

## C.6 STATISTICAL ANALYSIS FOR ALTERNATIVE EMISSION LIMITS

In selecting the numerical emission limits for each fuel type, it was necessary to determine the tradeoffs associated with standards of performance based on each of the three alternatives of applying a 40 percent $NO_x$ emissions reduction. Thus, a statistical approach was considered the most logical approach.

The uncontrolled $NO_x$ emission from each engine fuel type were assumed to follow a theoretical normal distribution. Figures C-35, C-36, and C-37 illustrate the theoretical normal distribution curves for diesel, dual-fuel, and gas engines, respectively. Note that the curves have truncated ends at positive points on the axis as no engines emit 0 g/hp-hr. The areas under each curve represent the population if each engine fuel type emitted up to a particular limit. The mean values which bisect the normal distribution were assumed to be the sales-weighted uncontrolled average $NO_x$ emission levels discussed in Section 4.3.4 and the standard deviations, for each fuel type were assumed to be equal to the standard deviation calculated from the data base in Appendix C. That is, the standard deviation is not sales weighted as no such information was available. If the standard deviation were calculated for increasingly large samples and increasing small class intervals, its value would be expected to approach the theoretical standard deviation. The data tabulated in Appendix C was used to calculate sample standard deviations for each fuel type: (1) Gas, $\sigma = 4$ g/hp-hr, (2) Dual-fuel, $\sigma = 3.2$ g/hp-hr, and (3) Diesel, $\sigma 3.7$ - g/hp-hr. As illustrated in the figures, about 68 percent of the engine population for a given fuel type will theoretically have an uncontrolled $NO_x$ emission level within plus or minus one standard deviation of the mean.

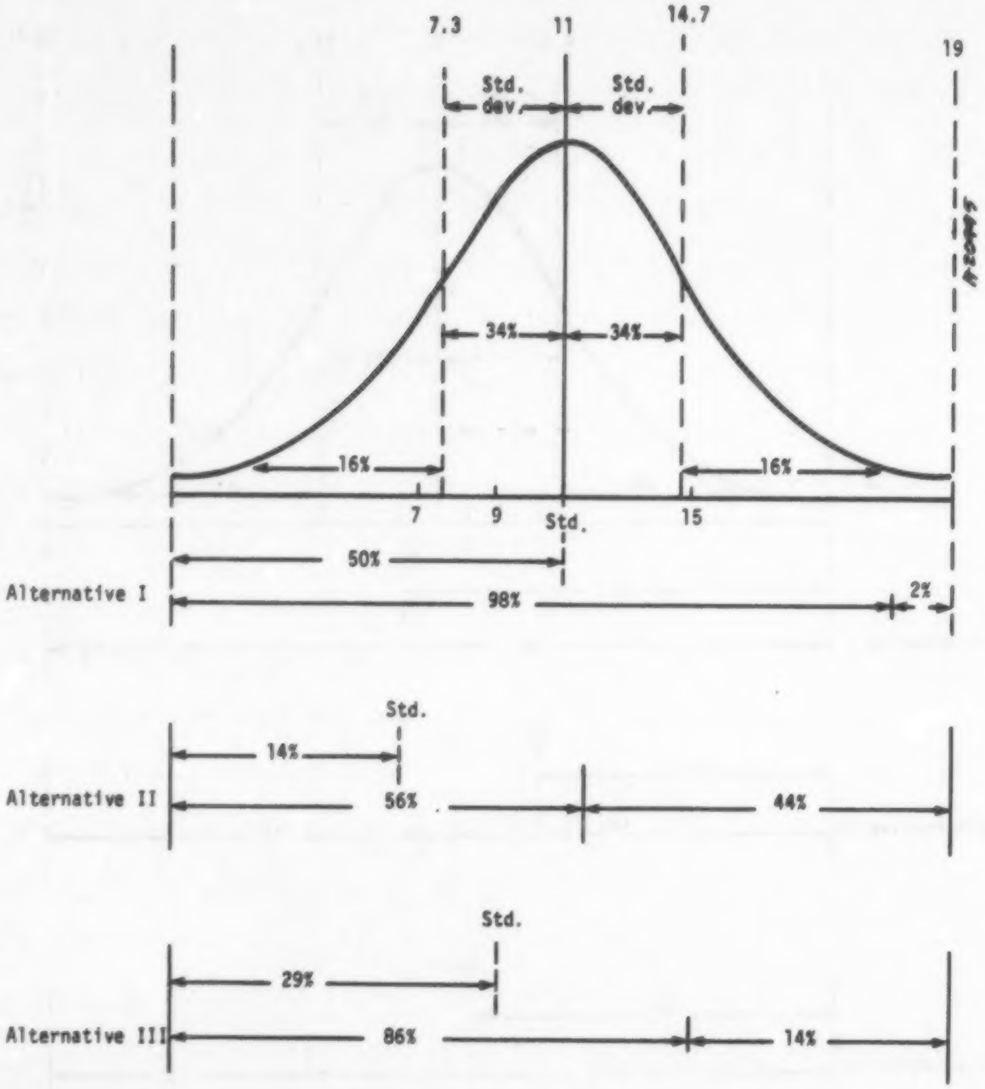


Figure C-35. Statistical effects of alternative emission limits on diesel engines.

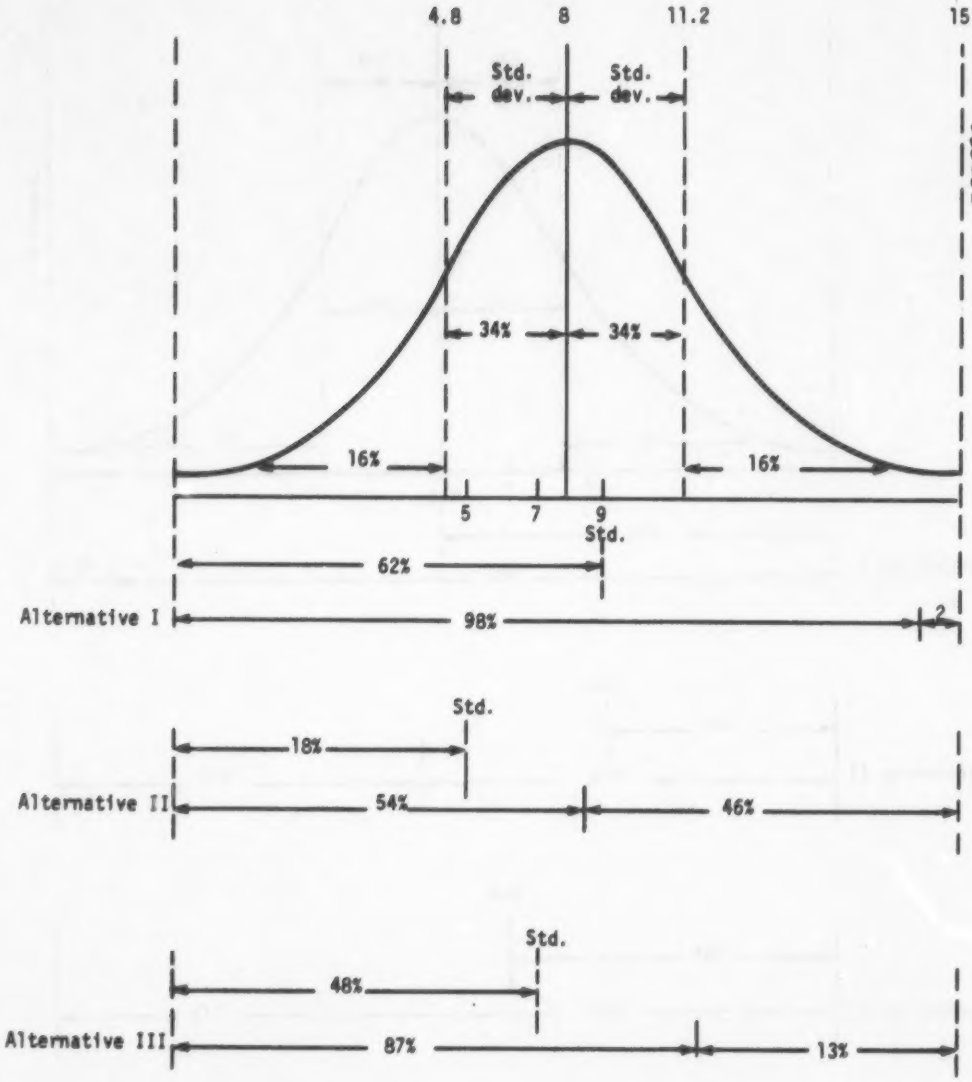


Figure C-36. Statistical effects of alternative emission limits on dual-fuel engines.

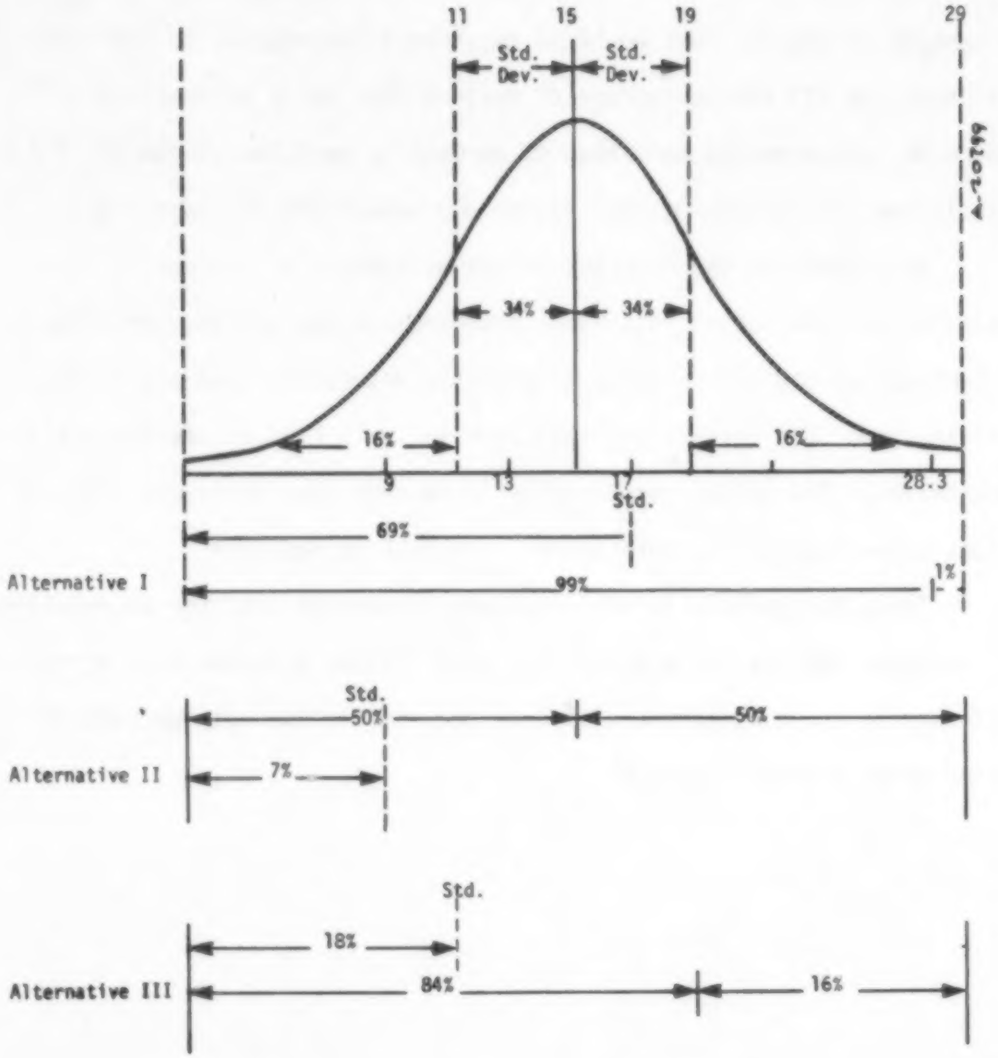


Figure C-37. Statistical effects of alternative emission limits on gas engines.

Standards of performance based on each of the three alternatives were analyzed relative to the theoretical normal distribution curves to determine: (1) the percentage of the engines that would have to reduce $NO_x$ emissions by 40 percent or less to meet the standard; (2) the percentage of engines that would be required to do nothing to meet the standard; and (3) the percentage of engines that would be required to reduce $NO_x$ emissions by more than 40 percent to meet the standard. The results are illustrated on each figure and summarized in Table C-93.

As a check on the accuracy of the assumption of a normal distribution, the actual data base in Appendix C was analyzed relative to percentages of engines for each alternative determined from the normal distribution. The results are tabluated in Table C-93 as the numbers in parentheses. The actual sample values show very good agreement with the values determined by the pure normal distribution approach.

Thus, it can be concluded that the assumption that the uncontrolled $NO_x$ emission levels, for a given fuel type, follow a theroetical normal distribution curve is essentially true and the statistical approach is deemed to be a valid approach.

## TABLE C-93. SUMMARY OF STATISTICAL ANALYSES OF ALTERNATE EMISSIONS LIMITS

### GAS ENGINES

| Alternative | I | II | III |
|---|---|---|---|
| Standard | 17 | 9 | 11 |
| Percent meeting standard with ≤40 percent control | 99 (100) | 50 (50) | 84 (91) |
| Percent required to do nothing to meet standard | 69 (67) | 7 (6) | 18 (9) |
| Percent required to apply 40 percent control to | 1 (0) | 50 (58) | 16 (9) |

### DUAL-FUEL ENGINES

| Alternative | I | II | III |
|---|---|---|---|
| Standard | 9 | 5 | 7 |
| Percent meeting standard with ≤40 percent control | 98 (100) | 54 (44) | 87 (89) |
| Percent required to do nothing to meet standard | 62 (67) | 18 (4) | 48 (44) |
| Percent required to apply 40 percent control to | 2 (0) | 46 (56) | 13 (11) |

### DIESEL ENGINES

| Alternative | I | II | III |
|---|---|---|---|
| Standard | 11 | 7 | 9 |
| Percent meeting standard with ≤ 40 percent control | 98 (100) | 56 (63) | 86 (90) |
| Percent required to do nothing to meet standard | 50 (65) | 14 (18) | 29 (45) |
| Percent required to apply 40 percent control to | 2 (0) | 44 (37) | 14 (10) |

## C.7 COMPARISON OF SMALL-BORE, AUTO ENGINE EMISSION CONTROL TECHNIQUES FOR LARGE BORE-STATIONARY IC ENGINES

### C.7.1 Introduction

During a review meeting with EPA in September 1977, several questions were raised concerning the application to large-bore stationary engines of control techniques that have been shown to be effective on automotive engines. In addition, questions were asked why catalytic reduction of $NO_x$ (which has been successfully used in Japan) and water induction (the primary $NO_x$ control technique for stationary gas turbines) were not alternative controls for stationary IC engines. This memo briefly summarizes these three questions.

### C.7.2.1 Comparison of Control Techniques for Large-Bore Stationary and Automotive Engines

Essentially the same $NO_x$ control techniques that have been used by engine manufacturers to meet emission regulations for mobile sources are also effective when applied to large-bore engines. These techniques include derating*, retarded ignition or fuel injection, manifold air cooling (with turbocharging), air-to-fuel ratio changes, and exhaust gas recirculation. Of these controls, retard (for diesel or dual fuel engines) and air-to-fuel changes (for natural gas engines) are particularly effective when applied to large-bore engines.

Nevertheless, inherent differences in the design and operating modes of large stationary and smaller, mobile engines have dictated different approaches to exhaust emission control. Figure C-38, which shows the effect of A/F ratio on exhaust emissions, can be used to illustrate this point. Typically, spark ignition automotive engines are operated at slightly less than stoichiometric A/F ratios. This operation is largely dictated by the difficulty in closely controlling the A/F ratios to individual cylinders within these carbureted engines. Therefore, A/F ratios must be maintained richer than stoichiometric to avoid detonation and misfiring.

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*Derating is accomplished by using a larger engine than necessary for a particular car.

With the advent of strict emision regulations (primarily aimed at reducing HC and CO emissions), automotive manufacturers concentrated on increasing the A/F ratio (e.g., "lean burn" engines), which, as Figure C-35 illustrates, decreases HC and CO emissions significantly and also improves fuel economy. $NO_x$ emissions, however, increase initially as A/F ratios are increased beyond stoichiometric. As emission regulations became even more stringent, additional controls were, therefore, required. These controls included exhaust gas recirculation (EGR) and returning to richer A/F ratios to reduce $NO_x$. To compensate for the resulting increase in HC and CO emissions, oxidizing catalysts were added. These catalysts also enabled automobiles to reach the more stringent requirements for HC and CO that became mandatory in 1975. Improvements in carburetion and mixing were also effective in optimizing exhaust emissions over a range of loads and speeds.

Lage bore engines, in contrast, typically operate leaner than stoichiometric. Consequently, as shown in Figure C-35, $NO_x$ emissions

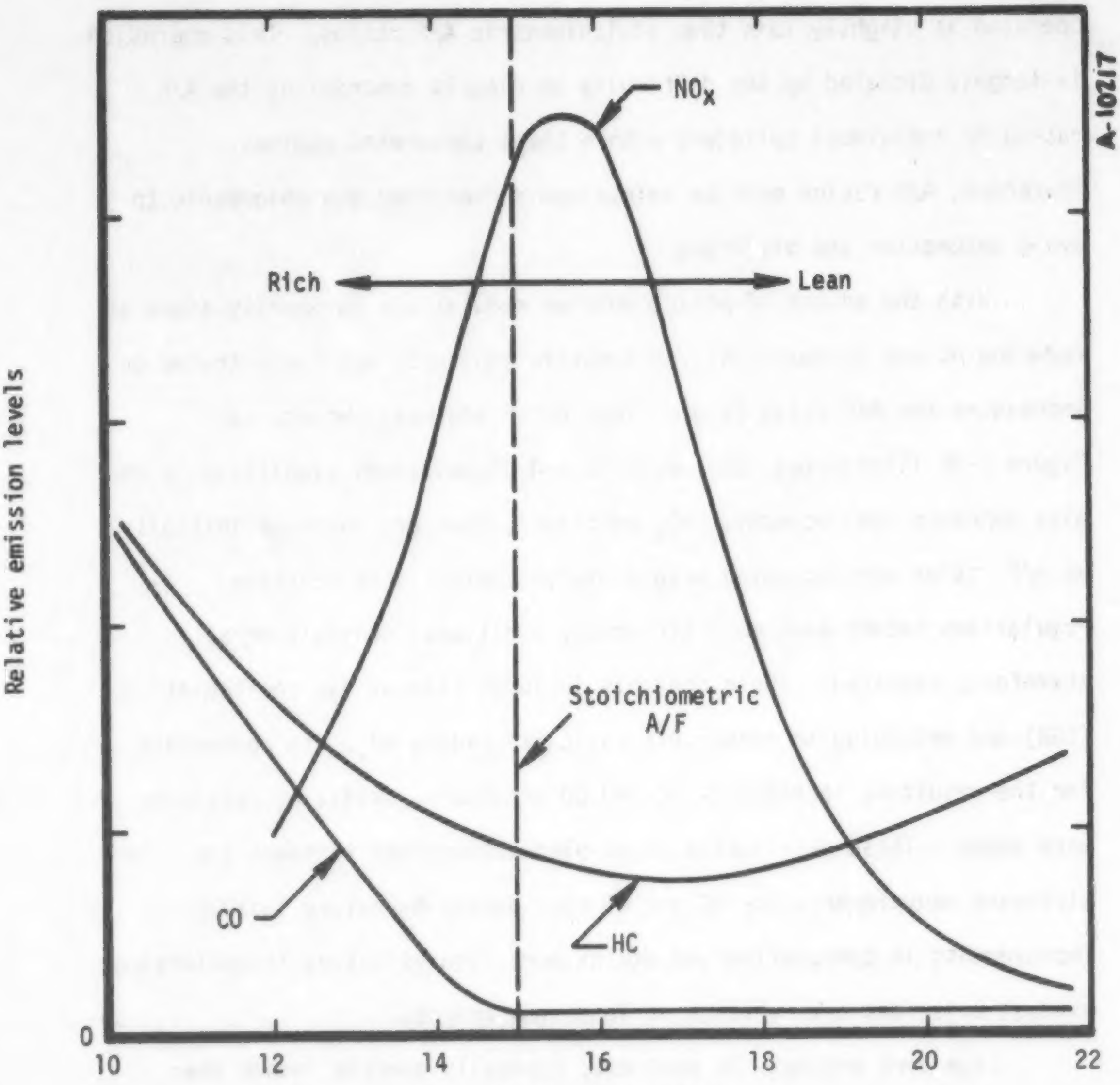

Figure C-38. Effect of A/F ratio on emissions from a gasoline engine.

are relatively high, but HC and CO emissions are now low*. These operating conditions result from the need to optimize fuel consumption of large-bore engines because they see high usage, high load service. Therefore, the emphasis of exhaust emission control for large-bore engines is in reducing $NO_x$ emissions. Although the same types of combustion modifications that are effective in reducing $NO_x$ from automotive engines could be used on large stationary engines, the selection of optimum controls and the direction and degree of application of the selected controls is different due to differences in uncontrolled A/F ratios and fuel charging methods.

### C.7.2.2 Catalytic Reduction

As of this date no large-bore engine manufacturers are using catalysts to reduce $NO_x$ emissions. $NO_x$ reduction catalysts, until 1977, have not been used on gasoline or diesel vehicles either. The catalysts that have been and are still currently in use on automobiles are primarily oxidizing catalysts, whose purpose is to lower HC and CO emissions. Recently, however, three-way catalysts have been developed for the reduction of HC, CO, and $NO_x$ emissions and the first commercial application was for a 1977 Volvo sold in California. (Three-Way Conversion Catalysts -- Part of the New Emission Control System; SAE Paper, 770365). These three-way catalysts are similar to the earlier oxidizing catalysts (precious metals coated on monoliths or pellets) with the addition of rhodium which selectively reduces $NO_x$ when the A/F ratio is maintained very near stoichiometric. These conditions exist in automotive exhausts but not in the discharge from large bore engines. Therefore, this control approach shows little promise for large bore engines.

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*The exceptions to this generalization are those large-bore engines which are naturally aspirated or carbureted; these units have emissions characteristics which are similar to automotive engines since they operate at A/F ratios closer to stoichiometric.

As discussed on page 4-139 of the SSEIS, however, an ammonia/catalyst $NO_x$ control system has been suggested in the literature and is based on the successful application of this technique to nitric acid plant tail gases and a number of large stationary combustion sources in Japan. An excellent discussion of technique is presented in a paper "Status of Flue Gas Treatment Technology for Control of $NO_x$ and Simultaneous Control of $SO_x$ and $NO_x$", Mobley, J. David, and Stern, Richard D., U.S. Environmental Protection Agency, Report No. EPA-600/7-77-033c. This paper is an excellent review of this technology, its effectiveness, and its costs. Based on information in this report it would appear that selective catalytic reduction is a technically feasible approach for gaseous fuel engines, but the fouling (particulate) and catalyst-poisoning (sulfur) problems associated with oil or coal combustion would require further development. Moreover, this approach appears to be very expensive. If the published costs are extrapolated for IC engines, the data indicate that annual ownership and operating costs would increase from between 3 to 30 percent. These costs are based on sources that are at least an order of magnitude larger than a typical engine installation. In addition, the costs of maintenance and catalyst replacement may be somewhat understated for smaller sources, such as IC engines. Thus, it is probable that the actual costs for an engine application would be even greater than these estimates.

A similar $NO_x$ reduction scheme has been developed by Exxon (Thermal Denox Process) for $NO_x$ reduction from stationary boilers and furnaces. In this process, ammonia is injected into the flue gas at locations where the temperature exceeds 1500°F. There appears to be no application for this process to IC engine exhausts, however, since exhaust temperatures are generally less than 1000°F.

### C.7.2.3 Water Induction

As discussed in the SSEIS (Section 4.4.7), water induction has been investigated by four large bore engine manufacturers (White Superior, Ingersoll-Rand, Cooper, and GMC/Electromotive). All reported serious concern about the feasibility of this technique based on observations of water in the crankcase and lubricating oil, rapid build-up of mineral scale (untreated water) around intake/exhaust valves and other components, and combustion deposits. All of the tests were of short duration (less than 25 hours of engine operation), and manufacturers believe much longer tests (2000 to 8000 hours) would be necessary to establish the effect of water induction on wear rates and operating reliability.

In addition to these manufacturers, two smaller bore engine manufacturers, GMC/Detroit Diesel and Caterpillar, have reported tests of water induction in truck-size diesel engines. Caterpillar believes the technique is viable if the water is treated for freeze protection and mineral content. Caterpillar tests showed significant deposits, but they concluded these could be prevented by demineralizing the water. GMC/Detroit Diesel also experienced significant deposits and concluded that these deposits originated from condensation of combustion products rather than water minerals. Based on their exprience, GMC/Detroit Diesel concluded that water induction was not feasible in their 2-stroke designs.

## C.8 Determination of Sales-Weighted Average Uncontrolled CO and HC Emissions

Since the source of variability caused by engine design cannot be specifically identified, a procedure is required to characterize uncontrolled CO and HC emissions levels of engines which are sold for similar applications.

The procedure adopted here is to compute a weighted, average uncontrolled emission level for engines in the diesel, dual-fuel, or natural gas categories. The three weighted levels are based on sales of engine horsepower during the past five years for domestic applications. Sales of horsepower to standby services were excluded from this computation, since engines sold for standby applications will be exempted from standards of performance (see Chapter 9).

The sales-weighted averages for diesel, dual-fuel, and natural gas engines are presented in Figures C-39(a-f), which also show each manufacturer's CO and $HC_t$ data. The weighted average uncontrolled CO level for diesel engines is 2.9 g/hp-hr; for dual-fuel units, 2.7 g/hp-hr; and for natural gas engines, 7.7 g/hp-hr. The weighted average uncontrolled $HC_t$ level for diesel engines is 0.3 g/hp-hr; for dual-fuel units, 2.8 g/hp-hr; and for natural gas engines, 1.8 g/hp-hr.

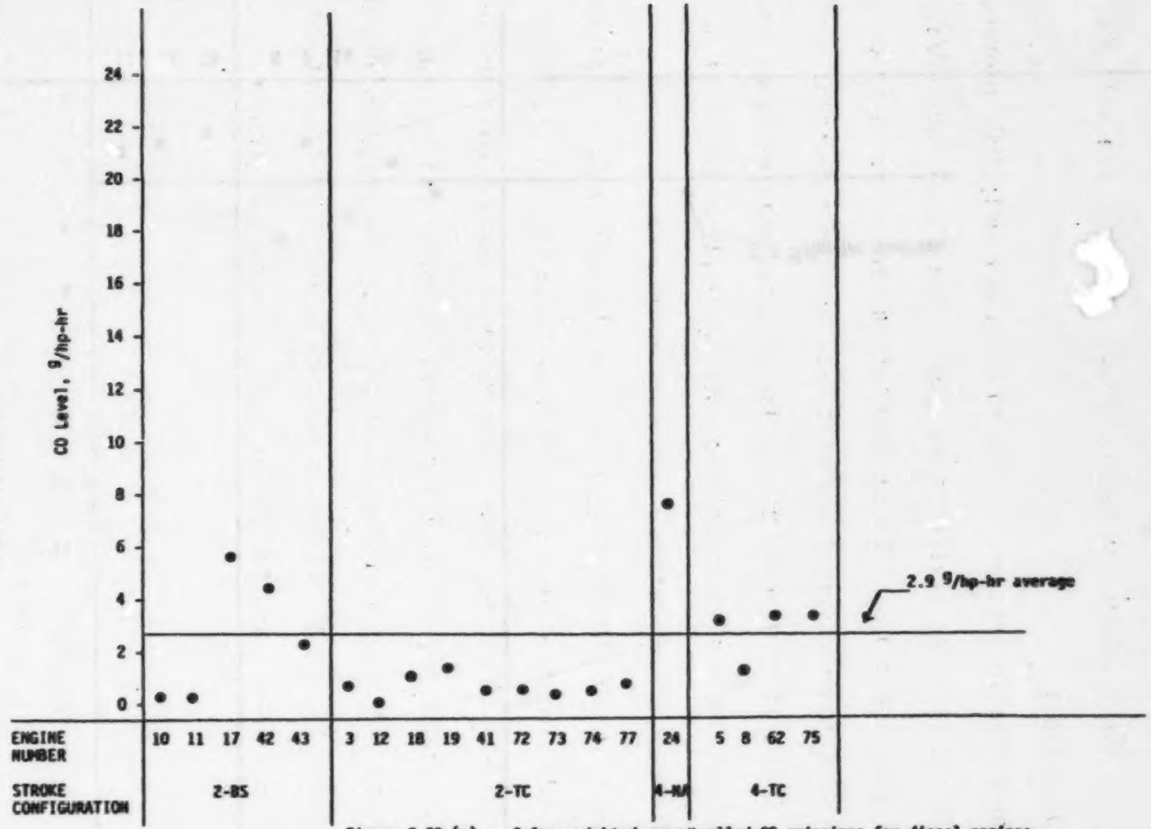

Figure C-39 (a). Sales-weighted uncontrolled CO emissions for diesel engines.

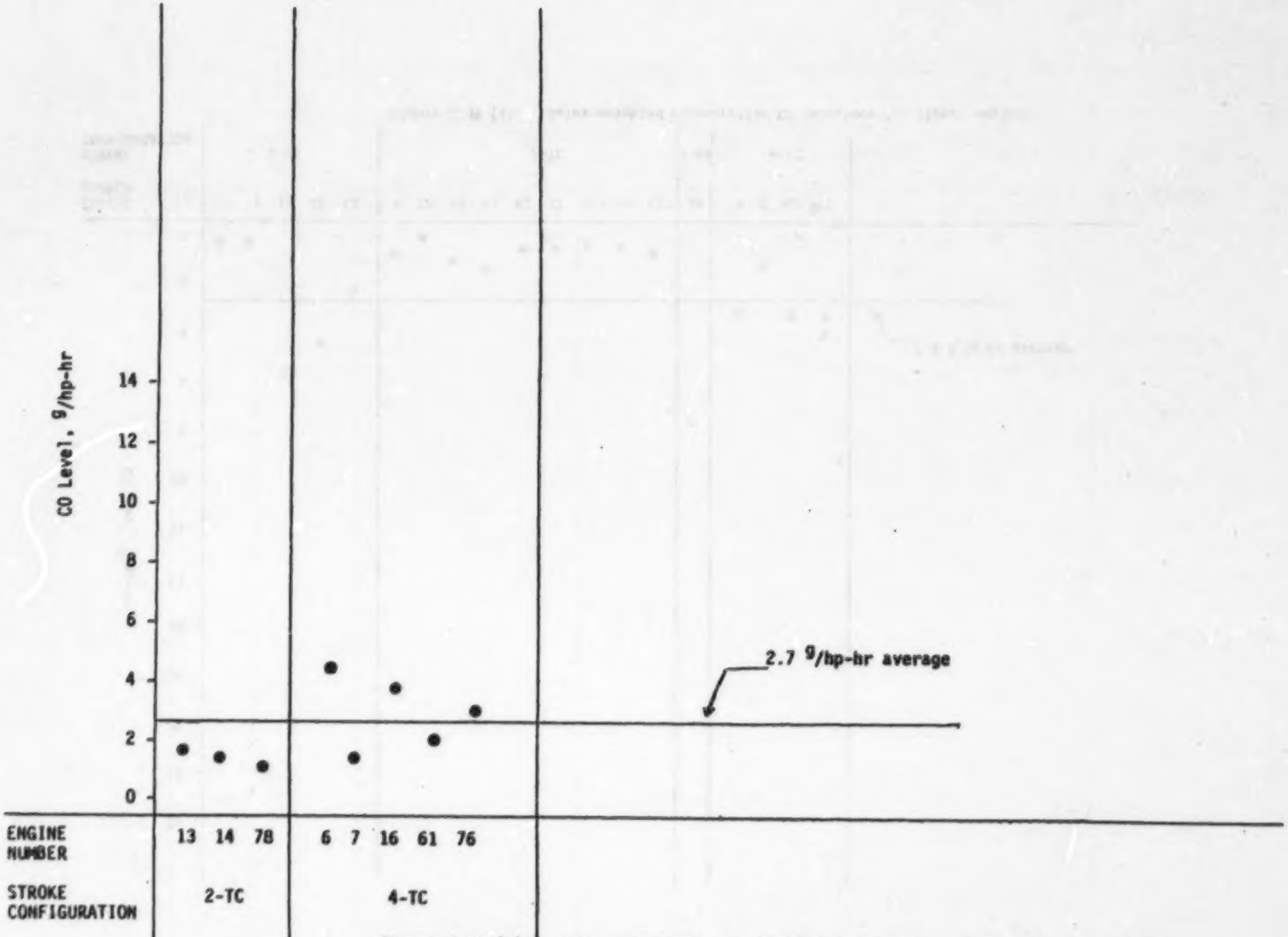

Figure C-39 (b). Sales-weighted uncontrolled CO emissions for dual-fuel engines.

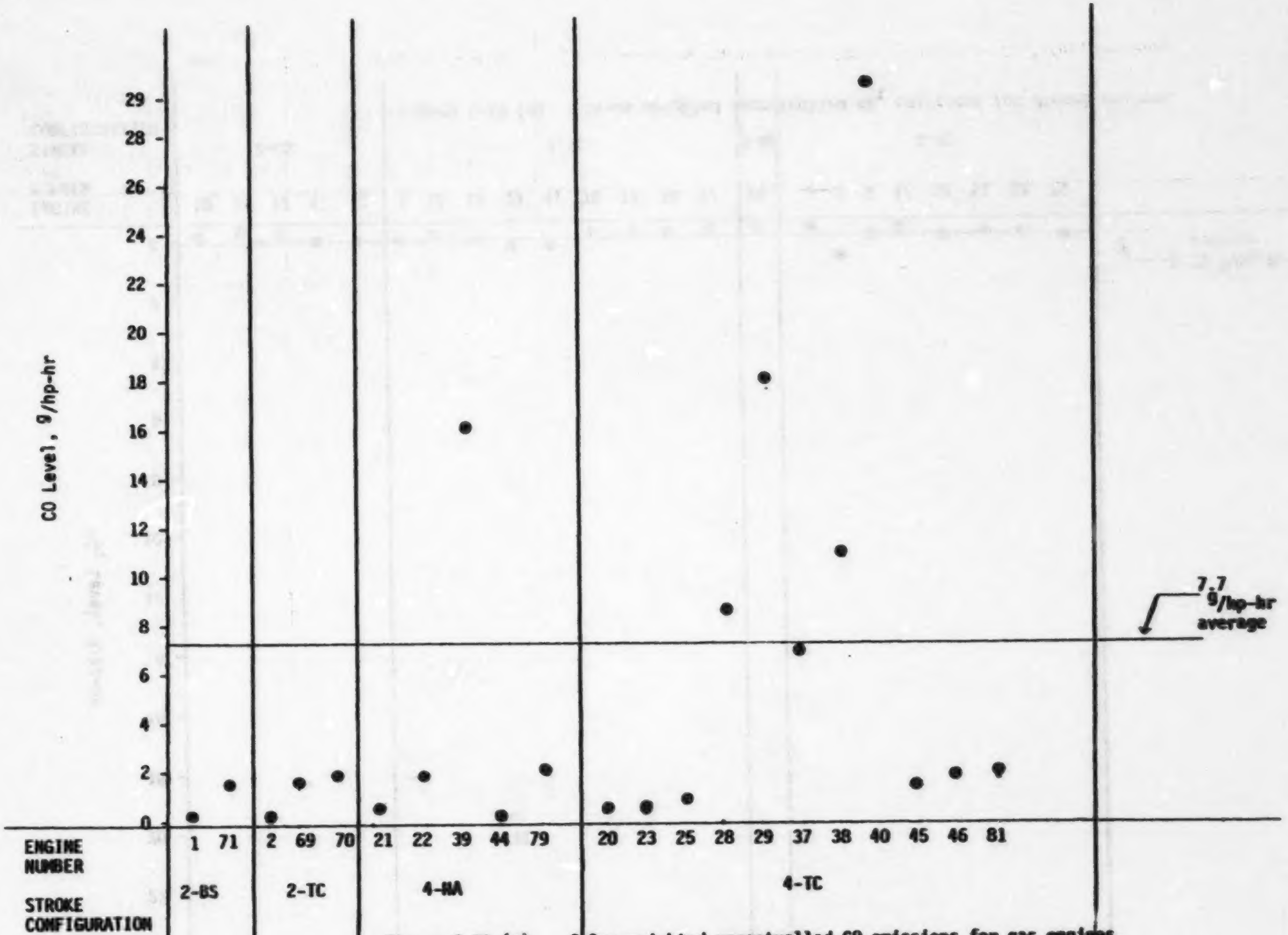

Figure C-39 (c). Sales-weighted uncontrolled CO emissions for gas engines.

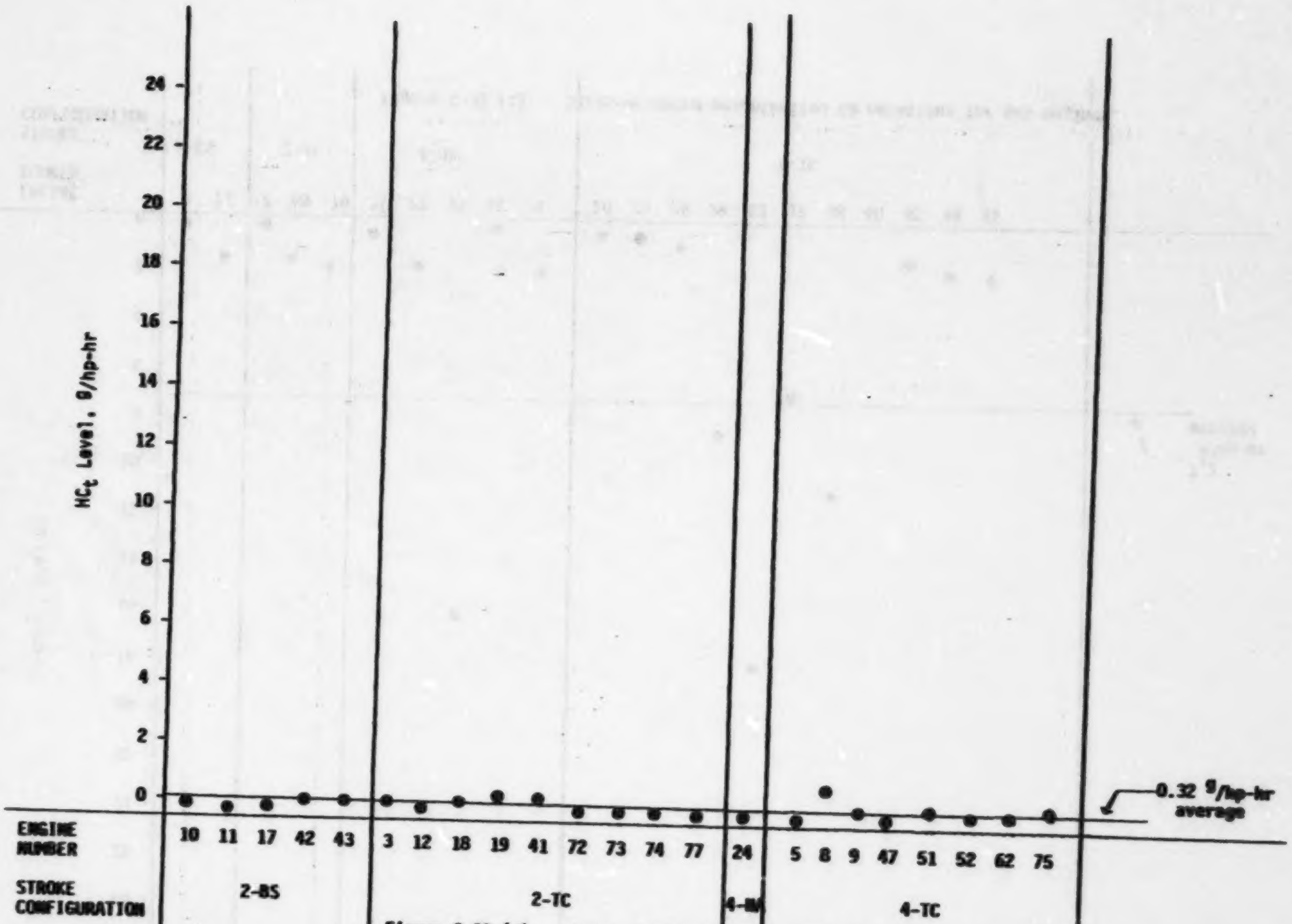

Figure C-39 (d). Sales weighted uncontrolled HC$_t$ emissions for diesel engines.

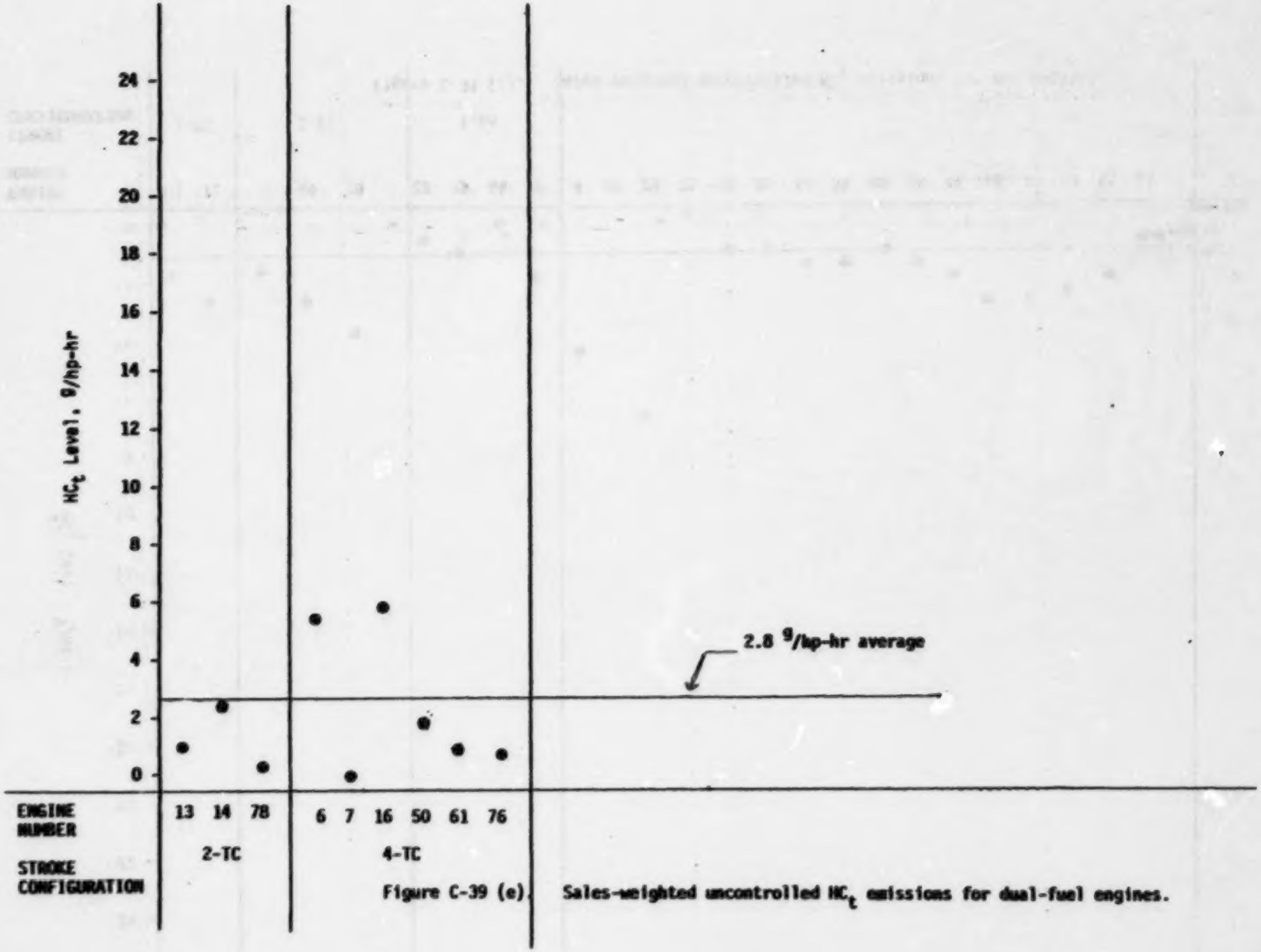

Figure C-39 (e). Sales-weighted uncontrolled HC$_t$ emissions for dual-fuel engines.

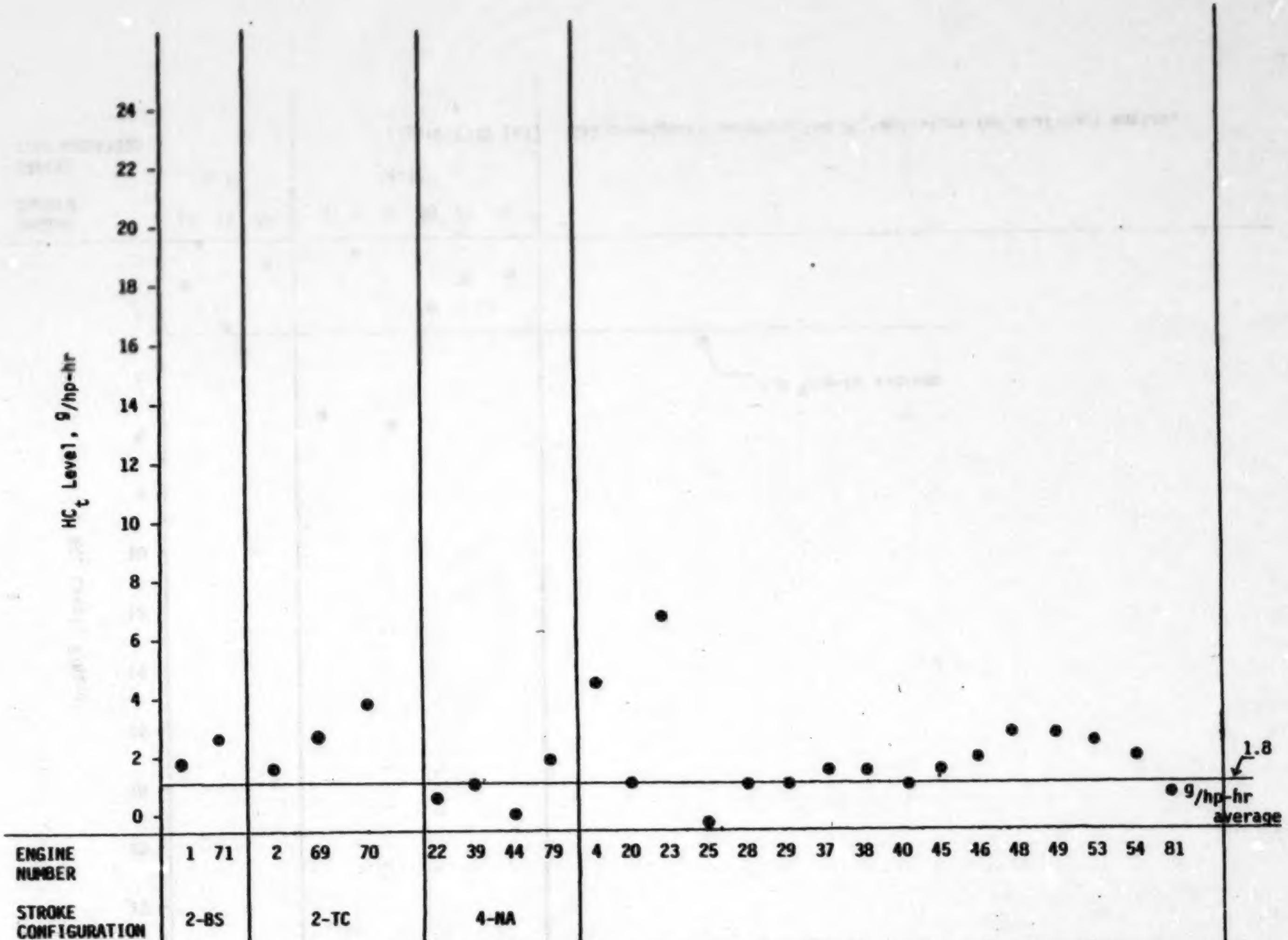

Figure C-39 (f). Sales-weighted uncontrolled HC$_t$ emissions for gas engines.

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## GLOSSARY

| | |
|---|---|
| aftercooler | -- a heat exchanger used to cool inlet air that has passed through a turbocharger. Also referred to as an intercooler (see Figure 4-32). |
| air-cooled | -- a method of engine cooling, often used on small engines. Air is sucked or forced over the engine (by an engine-driven fan) transferring heat from the engine block. Fins are generally used to conduct heat away from the combustion chambers and increase the area available for heat transfer. |
| air injection | -- an emission control technique in which air is pumped (injected) into the exhaust manifold to complete the reaction of any unburned fuel and CO which is still present. |
| air-to-fuel ratio | -- a ratio of the mass flowrate (g/hr) of air into an engine to the mass flowrate of fuel (g/hr). |
| annualized cost | -- initial costs, or the sum of initial and operating costs, expressed as an annual charge. |
| baseload | -- continuous operation, generally considered to be 8000 hr/yr. |
| BDC | -- see bottom dead center. |
| blower scavenging | -- a method of charging the cylinder of an engine with air in which a low-pressure blower driven by the engine forces air into the cylinder. |
| bore | -- the diameter of the cylinder of an engine. |
| bottom dead center | -- position of the piston when it is at the bottom of the cylinder; corresponds to maximum available gas volume. |
| brake horsepower (bhp) | -- the power delivered by the engine shaft at the output end. The name is derived from the fact that it was first measured by the power consumed in a brake attached to the output shaft. Brake horsepower equals |

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| | total power delivered by the pistons less losses and power used to drive auxiliaries (fan, pumps, etc.). |
| brake mean effective pressure (bmep) | -- the hypothetical constant pressure that would have to be exerted on the head of the piston during the entire stroke to generate the same torque as is actually generated by the engine. Hence, frequently used synonymously with torque. |
| brake specific | -- emissions expressed on the basis of power output, i.e., emissions g/hp-hr. |
| carburetor | -- a device on a spark ignition engine which controls the flowrate of air and fuel and mixes them in the proper proportions for combustion. |
| catalytic converter | -- a device which uses a catalyst to promote a reaction that alters the chemical composition of the gas passing through it. Oxidation type converters change CO and HC to $CO_2$ and $H_2O$ using a precious metal catalyst, such as platinum. Reduction catalysts are used to reduce NO to $N_2$ and $O_2$. |
| cetane number | -- a reference number for compression ignition engine fuels. Higher numbers indicate better ignitability and better antiknock characteristics. |
| chemiluminescent analyzer | -- a device used to measure $NO_x$ emissions. |
| compression ignition (CI) | -- one of two methods of initiating combustion in the engine cylinder. In CI engines, the air charge is introduced into the cylinder and compressed, thereby raising its temperature above the auto-ignition temperature of the fuel (temperature at which the fuel ignites spontaneously). Fuel is then injected into this hot compressed air and ignites spontaneously. All diesel and dual fuel engines are compression ignited. |
| compression ratio (CR) | -- ratio of the volume in the combustion chamber when the piston is at the bottom of the stroke to that when the piston is at the top. |
| connecting rod | -- a rod which connects the piston to the crankshaft and permits the reciprocating motion of the piston to be transferred to rotary motion by the crankshaft. |
| continuous rating | -- see rating. |
| cooling system | -- the system by which combustion heat is removed from engine block. This system may consist of a water jacket through which a liquid is circulated to remove |

heat from the engine and a radiator, cooling tower, or heat recovery system to remove the heat from the liquid before it is recirculated through the water jacket. It may also consist of fins to conduct heat away from the combustion chambers and a fan to move air past these fins to remove the heat from them (see "air-cooled").

cost effectiveness -- ratio of the cost to reduce a pollutant from an engine to the amount of pollutant removed ($/kg).

crankcase blowby -- unburned fuel, combustion gases, and lubricating oil which escape from the cylinder past the piston rings into the crankcase and are then vented.

crankshaft -- the shaft that receives engine power from the reciprocating motion of the pistons and delivers it as rotary motion.

crude -- unrefined oil.

cylinder liner -- a steel liner inserted into the cylinder. It can be removed and replaced after excessive wear.

denitrification -- the removal of nitrogen from fuel during refining. Some removal occurs concurrently with sulfur removal.

derating -- a control technique which limits the maximum load of an engine to less then the design value.

desulfurization -- the removal of sulfur from fuel. This occurs during the refining of crude oil into light distillates and gasoline.

digester gas -- fuel gas formed from sewage sludge. Primary constituents are $CH_4$ (methane), $CO_2$, and $H_2$.

direct injection -- the injection of fuel directly into the cylinder.

displacement -- the volume a piston sweeps out moving from the bottom of the cylinder (bottom dead center, BDC) to the top of the cylinder (top dead center, TDC).

distillate -- a product of the refining of crude oil. The most common distillates are No. 2 diesel oil and gasoline.

dribble -- the loss of fuel from the injector tip after fuel injection.

dry -- refers to gas measurements made under conditions where water has been removed from the gas before the measure is made. Also gas measurements which have been

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| | mathematically adjusted to be equivalent to measurements made on water-free gas. |
| durability | -- the ability of an engine to operate throughout a normal service life without excessive wear or failure of engine components. |
| exhaust gas recirculation (EGR) | -- an emission control remote technique whereby a portion of the exhaust gases are retained in the cylinder (internal EGR) or are routed back to the engine intake (external EGR) to displace some of the inlet air. |
| exhaust manifold | -- an internally ducted casting which receives exhaust products from the cylinders and transfers these products to the exhaust systems (see Figure 4-47). |
| field gathering | -- collection of oil or natural gas at the surface of a well (the well-head) into the pipes (feeder lines) which carry it to the major pipelines. |
| flame ionization detector (FID) | -- an analytical instrument used to measure HC emissions. |
| fossil steam | -- steam for electric utility power generation produced from the combustion of coal, oil, or natural gas. |
| four-stroke | -- a type of engine which requires four traverses of the piston in the cylinder (two revolutions of the crankshaft) per power stroke (i.e., to complete a cycle). |
| fuel additive | -- a substance added to fuel, usually to reduce smoke from an engine. |
| fuel-bound nitrogen | -- nitrogen contained in the fuel rather than in the air. |
| fuel pump | -- device which pumps fuel to the fuel injection system or carburetor. |
| governor | -- device which controls the amount of fuel supplied to the cylinder according to the load demand on the engine, e.g., a governor is used to maintain a given speed (rpm) under varying load in electric generator applications. |
| heavy duty mobile | -- a term referring to vehicles over 6000-lbs gross vehicle weight (total weight when vehicle is fully loaded) subject to federal emission standards. |
| higher heating value | -- the heat produced by the complete combustion of a unit quantity of fuel (at standard conditions of temperature, pressure, and humidity) such that the |

water in the products of combustion is in the liquid phase.

**horsepower** -- the time rate of doing work. One U.S. horsepower is equal to 33,000 foot-pounds per minute. (One U.S. horsepower equals 1.014 metric horsepower.)

**indirect injection** -- the injection of fuel into a precombustion chamber, or antechamber, where combustion commences in an oxygen deficient environment before expanding into the cylinder for completion of the combustion in the presence of excess oxygen (see Figure 4-42).

**initial cost** -- the purchase price of the engine, including all auxiliaries necessary for its operation (e.g., radiator, lube oil pump, aftercooler, starting motor, mounting skid, etc.).

**injection rate** -- the rate at which fuel is introduced into the cylinder by the injector.

**injection timing** -- the time, measured in degrees of crankshaft rotation, that it takes for fuel to be admitted into the cylinder. Start of injection (in degrees before TDC) is an important parameter in emissions control.

**injector** -- device which injects fuel into the cylinder (see Figure 4-55).

**injector rack** -- a mechanical linkage controlled by the governor which determines the pressure of the fuel supplied by the fuel pump to the fuel injector.

**intake manifold** -- internally ducted casting that distributes incoming air or air and fuel mixture into the cylinders (see Figure 4-32).

**intercooler** -- see aftercooler.

**intermittant rating** -- see rating.

**jacket water** -- engine cooling water that circulates from a cooling tower or radiator through cavities (or jackets) in the engine block.

**knock** -- premature ignition of fuel in cylinder of an engine.

**lean** -- refers to an air and fuel mixture which contains more air than stoichiometrically necessary to completely burn all the fuel.

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| low sac nozzle injector | -- a fuel injector with a minimum of volume beneath the injector nozzle, thus reducing injector dribble and HC emissions (see Figure 4-55). |
| lower heating value | -- the heat produced by the complete combustion of a unit quantity of fuel (at standard conditions of temperature, pressure, and humidity) such that the water in the products of combustion is in the vapor phase. |
| major overhaul | -- an overhaul which generally includes removal and replacement (or reconditioning) of the cylinder liners, pistons, rings, valves, fuel pump, and crankshaft. The engine is frequently removed from the installation to perform this overhaul. |
| manifold air temperature cooling (MAT) | -- a control technique based on lowering the intake air temperature. Usually accomplished with an intercooler (see Figure 4-32). |
| maximum rating | -- see rating. |
| minor overhaul | -- an overhaul which generaly includes replacing rings, valves, injectors or spark plugs, and occasionally pistons. This overhaul is usually performed without removing the engine from its installation and at more frequent intervals than major overhauls. |
| naturally aspirated | -- a type of air charging whereby the piston draws air into the cylinder as it travels to the bottom of the cylinder. Not turbocharged, supercharged, or blower-scavenged. |
| nondispersive infrared analyzer (NDIR) | -- an instrument used to measure CO and $CO_2$. |
| nonpropulsive | -- an application where the engine is never used to move the structure or device on which it is mounted. |
| nuclear steam | -- steam (usually for electric generation) produced by heat from nuclear reactions. |
| operational change | -- a change in engine operation which requires only an adjustment of given operating conditions but no hardware additions -- e.g., ignition timing. |
| original equipment manufacturer (OEM) | -- a firm which, in this case, buys an engine from an engine manufacturer and incorporates it into a product of which the engine is only a component -- e.g., a portable compressor, which is usually assembled by an |

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| | OEM using his own housing and compressor but a purchased engine. |
| overhaul | -- the replacement of worn parts in an engine, usually classified as either a minor or major overhaul depending upon how many, and which, parts are replaced. (See minor and major overhaul.) |
| oxidation | -- when used in this document, oxidation refers to the addition of oxygen to a molecule by a chemical reaction, as when CO combines with oxygen to form $CO_2$. (The more general definition is the loss of electrons by a reactant in a chemical reaction.) |
| paramagnetic analyzer | -- an instrument used to measure oxygen content of exhaust. |
| peak cylinder temperature | -- the maximum temperature in the cylinder during combustion. |
| pilot oil | -- a small amount (<10 percent by heating value) of diesel fuel injected near the end of the compression stroke of a dual-fuel engine to ignite the gaseous fuel. |
| piston crown | -- the upper surface of the piston. |
| piston ring | -- a thin ring of metal which fits into a groove in the piston. Piston rings seal the combustion volume by filling the gap between the piston and the liner. |
| ports | -- openings in the cylinder liner through which air and fuel enter the cylinder and/or combustion gases leave it. Used in two-stroke engines. |
| precombustion chamber | -- see indirect injection. |
| physical change | -- a hardware change to an engine (as opposed to an operational change). |
| quenching | -- the cooling below the combustion point of fuel which has impinged on the walls of the cylinder. |
| rating | -- brake horsepower output of the engine. Reported either as continuous (power that engine can deliver continuously), intermittant (power that engine can deliver intermittantly, usually for no more than 1 hour during a consecutive 2-hour period), or maximum (peak power than engine can deliver). Continuous ratings are used in this document. |

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| reactivity (photochemical) | -- potential of a hydrocarbon compound to react with other species in the atmosphere to produce smog. |
| reduction | -- when used in this document, reduction refers to the removal of oxygen from a molecule by a chemical reaction, as when NO is reduced to $N_2$ and $O_2$ in the presence of a catalyst. (The more general definition is the gain of electrons by a reactant in a chemical reaction.) |
| residence time | -- the time interval after ignition during which the air-fuel mixture can burn at elevated temperature and pressure. |
| residual | -- a heavy viscous oil, often containing large amounts of sulfur (>1 percent by volume), which remains after distillation to produce other fuels. |
| retard | -- a $NO_x$ control technique wherein ignition of the fuel is delayed by delaying the spark (SI engines) or the start of fuel injection (CI engines) by several degrees of crankshaft rotation. |
| rich | -- refers to an air and fuel mixture which contains less oxygen than is stoichiometrically necessary to completely burn all the fuel. |
| rpm | -- rotations per minute; a measure of the engine crankshaft rotational speed. |
| sac | -- the small volume below the nozzle of an injector (see Figure 4-55). |
| scrubber | -- a device which removes a pollutant from an exhaust gas stream through absorption of the pollutant by the scrubber liquid. |
| smoke | -- visible emissions from an engine exhaust. |
| spark ignition (SI) | -- one of two methods of initiating combustion in the engine cylinders (see also compression ignition). In SI engines, an electrical spark is generated across a gap between two electrodes at the tip of the spark plug to ignite the fuel-air mixture. All gasoline and natural gas fueled engines are spark ignited. |
| spark timing | -- the degrees of crankshaft rotation before top dead center (TDC) at which the spark commences. |
| squish lip | -- refers to a cavity shape in the piston head which generates squish, or radially outward motion of the air-fuel mixture. Combustion is initiated in this cavity. |

**standby** -- a limited usage service, typically 200 hr/yr or less (e.g., emergency electrical generators).

**stationary** -- an application in which the engine is never used to propel the structure or device on which it is mounted.

**stoichiometric** -- refers to the chemically correct amount of air (oxygen) to completely burn a given amount of fuel.

**stratified charge** -- the staged combustion of fuel, first partially under rich conditions, and completely later at cooled temperatures in the presence of excess air (see Figures 4-44(a) and 4-44(b)).

**stroke** -- the vertical distance that the piston travels between the bottom and the top of the cylinder. Also the movement of the piston between these points.

**swirl** -- rotational motion of the air or air-fuel mixture in the cylinder. Used mainly to enhance air-fuel mixing. Can be low, medium, or high swirl depending on ratio of rotational speed to average piston speed.

**TDC** -- see top dead center.

**thermal efficiency** -- ratio of the engine energy output to the energy input based on the lower heating value of the fuel used.

**thermal fixation** -- the formation of $NO_x$ by reaction of nitrogen in the inlet air with oxygen during the combustion process.

**thermal reactor** -- a device which combines air with the unburned fuel and CO in the exhaust to promote further combustion of these gases, thereby reducing HC and CO emission.

**top dead center** -- position of the piston when it is at top of the cylinder; corresponds to minimum available gas volume.

**torque** -- a movement (the product of a force and a lever arm) which produces rotation.

**turbocharger** -- a device which uses a centrifugal compressor to increase the pressure of the incoming air. The compressor is powered by an exhaust gas driven turbine. It is used to increase the power output and efficiency of the engine (see Figure 4-32).

**two-stroke** -- a type of engine which requires two traverses of the piston in the cylinder (one revolution of the crankshaft) per power stroke (i.e., to complete a cycle).

| | |
|---|---|
| valve | -- a device which opens into the cylinder to admit air or air and fuel mixture or to exhaust combustion products. |
| valve camshaft | -- a shaft driven off the crankshaft having eccentric lobes which open the intake and exhaust valves at the proper time. |
| valve overlap | -- the interval, in degrees of crankshaft rotation, during which the intake and exhaust valve are both open. |
| volatility | -- a measure of the ability of a fuel to evaporate at a given temperature. |
| wet | -- refers to the existence of water vapor in an exhaust gas sample from water of combustion and water contained in the intake air. |

## TECHNICAL REPORT DATA

*(Please read Instructions on the reverse before completing)*

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**15. SUPPLEMENTARY NOTES**

This report discusses the proposed standards and the resulting environmental and economic impacts.

**16. ABSTRACT**

Standards of performance to control nitrogen oxides from new, modified and reconstructed stationary internal combustion engines in the U. S. are being proposed under section 111 of the Clean Air Act. This document contains information on the internal combustion engine industry and emission control technology, a discussion of the selected emission limits and the supporting data and the alternatives which were considered, and analyses of the environmental and economic impacts of the proposed standards.

**17. KEY WORDS AND DOCUMENT ANALYSIS**

| a. DESCRIPTORS | b. IDENTIFIERS/OPEN ENDED TERMS | c. COSATI Field/Group |
|---|---|---|
| Internal Combustion engines Air Pollution Control equipment Standards of Performance Pollution Control. | Air pollution control sulfur dioxide Nitrogen Oxides Water injection. | |

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